



**INFLUENCE OF COOLANT FLOW RATE PARAMETERS IN SCALING GAS
TURBINE COOLING EFFECTIVENESS ON A FLAT PLATE**

THESIS

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AFIT-ENY-MS-18-M-256

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Presented to the Faculty
Department of Aeronautics and Astronautics
Graduate School of Engineering and Management
Air Force Institute of Technology
Air University
Air Education and Training Command
in Partial Fulfillment of the Requirements for the
Degree of Master of Science in Aeronautical Engineering

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March 2018

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Abstract

In order to increase the efficiency of modern gas turbine engines, engine designers often set the gas temperature leaving the combustor section at a higher temperature than the melting point of the components downstream. This requires the turbine components to be cooled so that they maintain their designed shape and structural integrity. One method of cooling, called film cooling, takes bleed air from the compressor section and pushes it through small holes in the turbine blade's surface to create a relatively cool buffer between the hot freestream gas and the blade. Effective film cooling relies on the performance of the selected cooling hole shape and placement of the holes on the surface of the blade. To test each hole's effectiveness, large scale, low-temperature testing has been used to reduce cost and complexity. The resulting quantity from this testing is the adiabatic wall temperature (T_{aw}), which is defined as the temperature that the wall would be if the surface was adiabatic. This temperature is then nondimensionalized by the freestream and coolant temperatures, and the resulting quantity is called the adiabatic effectiveness (η). Because it is non-dimensional, the adiabatic effectiveness can then be scaled up to engine operating conditions to predict how the selected film cooling hole will perform. This process is complicated by the nonlinear variation of gas properties between coolant and freestream conditions, as well as between the experimental and engine conditions. The majority of previous scaling research focused on the effects of the density ratio (DR) between the coolant and freestream gases, ignoring other gas properties such as the specific heat. In this study, various gases were used as coolant to simulate changes in gas properties as one might have between experimental and engine conditions. It was found that when specific heat is included in the form of a new parameter, known as the Advective Capacity Ratio (ACR), adiabatic effectiveness results scale well be-

tween different cooling gases. The results indicate that ACR may be an appropriate parameter under certain conditions such that low temperatures experimental results can accurately be scaled up to engine conditions. Specifically at low momentum flux ratios, use of ACR produced very similar adiabatic effectiveness profiles between all gases, a result that has been rare in the literature using previously defined parameters.

Acknowledgements

I would like to thank my advisor, Lt Col James Rutledge, for the hours of advice and guidance throughout the course of this project. I would not have been able to complete this thesis without his help. I would also like to thank Dr. Marc Polanka for his help editing and shaping my Chapters 2 and 3, as well as for pushing me to become a better presenter. I would also like to thank 1st Lt Carol Bryant for always being available whenever I had a question or challenge in the lab. I wish to thank Dr. Andy Lethander and Natalia Posada of AFRL/RQTT, without whom this research could not take place. Finally, I would like to thank my wife, for her unwavering support and being there to give me that extra push when I needed it most.

Jacob P. Fischer

Table of Contents

	Page
Abstract	i
Acknowledgements	iii
List of Figures	vi
List of Tables	x
Nomenclature	xi
I. Introduction	1
II. Literature Review	5
2.1 Film Cooling Overview	5
2.1.1 Hole Shape Effects	6
2.1.2 External Flow Effects	11
2.2 Measurement Techniques	12
2.2.1 Thermal Methods	12
2.2.2 Mass-Transfer Methods	15
2.3 Scaling	17
2.3.1 Density Effects	18
2.3.2 Temperature Separation Effects	22
2.3.3 Specific Heat Effects	24
III. Methodology	29
3.1 Facility	29
3.2 Model Design	31
3.3 Experimental Methods	38
3.3.1 Data Acquisition and Reduction	41
3.4 Uncertainty Analysis and Calibration	43
3.4.1 Freestream, Coolant, and Gas Property Uncertainties	43
3.4.2 IR Camera Calibration	44
3.4.3 Adiabatic Effectiveness Uncertainty	47
IV. Results and Discussion	49
4.1 Temperature Separation Results	49
4.2 Adiabatic Effectiveness Results	51
4.2.1 Blowing Ratio, M	52
4.2.2 Momentum Flux Ratio, I	55

	Page
4.2.3 Advective Capacity Ratio, ACR	58
4.2.4 Velocity Ratio, VR	62
4.2.5 Reynolds Number Ratio, ReR	64
4.3 Analysis of Previously Collected Data	69
V. Conclusion	72
5.1 Conclusions	72
5.1.1 Temperature Separation Results	72
5.1.2 Scaling Results	73
5.2 Significance of Research	74
5.3 Recommendations for Future Work	74
Bibliography	76

List of Figures

Figure		Page
1.1.	Brayton cycle with engine station numbers for a typical gas turbine engine.	2
2.2.	Three hole geometries: a) Cylindrical b) Fan-Shaped c) Laid-Back Fan-Shaped [7].	7
2.1.	Adiabatic effectiveness values for a cylindrical hole at various blowing ratios [2].	8
2.3.	Adiabatic effectiveness distributions for the three hole geometries pictured in in Figure 2.2 [7].	8
2.4.	Four different cooling hole layouts used by Dittmar et al. [4].	9
2.5.	Adiabatic effectiveness for five different test configurations at a low blowing ratio [4].	10
2.6.	7-7-7 hole design parameters and layout [13].	10
2.7.	Effect of Reynolds number on laterally-averaged η [1].	12
2.8.	Difference in η_0 values calculated with 1D and 3D conduction assumptions [9].	14
2.9.	Structure of both a) single-component PSP and b) binary PSP [15].	16
2.10.	Resulting η_{max} values at all x/d locations from thermal and PSP measurements ($I = 1.0$) [15].	17
2.11.	Matched I , M , and VR cases with varying DR . Adapted from [14].	19
2.12.	η distributions at matched M and I with varying DR . Distributions from $x/D = 3$ and 15 are plotted. Adapted from [6].	20
2.13.	$\bar{\eta}$ distributions from three matched M values using air and CO_2 as coolant. Adapted from [5].	21
2.14.	Area-averaged η at various M and I values. Averaged over $x/D = 3-35$. Adapted from [13].	22

Figure	Page
2.15. ΔT distribution for He at $I = 0.25$, ($T_\infty - T_c = 0.1K$). Adapted from [15].	23
2.16. η distributions matching various coolant flow rate parameters [12].	25
2.17. η distributions matching $I = 0.5$ and 1.0 at $x/d = 5.0$ [16].	26
2.18. η distributions matching M and VR at 0.5 and 1.0 at the location $x/d = 5$. Adapted from [16].	27
2.19. η distributions at $x/d = 5$, matching ACR using IR and PSP methods. Adapted from [15].	27
2.20. η distributions for multiple matched ACR values for both air and CO_2 [15].	28
3.1. Wind tunnel schematic.	29
3.2. Coolant flow path schematic.	30
3.3. Test section schematic. Adapted from [15].	31
3.4. Top-down view of the new rig installed in the test section.	32
3.5. CAD model of the flat plate testing rig.	32
3.6. First plenum constructed for the single hole test insert, mounted behind test surface.	34
3.7. Raw IR image showing the coolant plume issues.	34
3.8. Second plenum constructed for the single hole test insert.	35
3.9. Test surface of two hole test insert with origin, spacing, and thermocouples indicated.	36
3.10. Plenums constructed for the two hole test insert.	37
3.11. Omega flowmeter used to monitor flow rates.	39
3.12. η_{app} distribution along $y/D = 0$ over the course of eight minutes.	40

Figure		Page
3.13.	Difference in η_{app} distributions after two minutes (left) and eight minutes (right).....	40
3.14.	Raw IR image of the spatial calibration set-up.	42
3.15.	Uncertainty in the coolant flow rate as a function of the reading [15].	44
3.16.	Generated IR calibration curve.....	46
3.17.	IR calibration curves from the flat plate and leading edge calibration rigs.	46
3.18.	IR calibration curves from two different Plexiglas wall temperatures.	47
4.1.	Difference between surface and freestream temperatures at various VR values using helium as coolant.	50
4.2.	Contours of η at the matched $M = 0.25$ case.....	52
4.3.	Distribution of η along $y/D= 0$ at the matched $M = 0.25$ case.	53
4.4.	Distribution of η along $y/D= 0$ at the matched $M = 1.50$ case.	54
4.5.	Area-averaged η values ($\bar{\eta}$) at all resulting M values. DR and $C_p R$ values listed are averages among all test points.	55
4.6.	Distribution of η along $y/D= 0$ at the matched $I = 0.25$ case.	56
4.7.	Distribution of η along $y/D= 0$ at the matched $I = 1.50$ case.	57
4.8.	Area-averaged η values ($\bar{\eta}$) at all resulting I values. DR and $C_p R$ values listed are averages among all test points.....	58
4.9.	Distribution of η along $y/D= 0$ at the matched $ACR = 0.25$ case.....	59
4.10.	Distribution of η along $y/D= 0$ at the matched $ACR = 0.50$ case.....	59

Figure	Page
4.11. Distribution of η along $y/D=0$ at the matched $ACR = 1.50$ case.	60
4.12. Area-averaged η values ($\bar{\eta}$) at all resulting ACR values. Deviation points at $I = 0.67$ labeled for each gas.	62
4.13. Distribution of η along $y/D=0$ at the matched $VR = 0.25$ case.	63
4.14. Distribution of η along $y/D=0$ at the matched $VR = 1.5$ case.	63
4.15. Area-averaged η values ($\bar{\eta}$) at all resulting VR values. DR and $C_p R$ values listed are averages among all test points.	64
4.16. Distribution of η along $y/D=0$ at the matched $ReR = 0.25$ case.	65
4.17. Distribution of η along $y/D=0$ at the matched $ReR = 1.5$ case.	66
4.18. Area-averaged η values ($\bar{\eta}$) at all resulting ReR values. DR and $C_p R$ values listed are averages among all test points.	66
4.19. Contours of η at the matched $ReR = 0.25$ case.	67
4.20. Contours of η at the matched $ReR = 1.5$ case.	67
4.21. Distribution of η along $x/D=3$ at the matched $ReR = 0.25$ case.	68
4.22. Distribution of η along $x/D=3$ at the matched $ReR = 1.5$ case.	68
4.23. Matched I cases with varying DR with calculated ACR values. Adapted from [14].	69
4.24. Matched M cases with varying DR with calculated ACR values. Adapted from [14].	70
4.25. Area-averaged η at various M and I values. Averaged over $x/D = 3-35$. Adapted from [13].	71

List of Tables

Table	Page
2.1. Properties of various gases for engine and testing conditions [8].	18
3.1. Steady state test conditions during data collection.	38
3.2. Uncertainties for the coolant gas properties [15].	43
3.3. Maximum and minimum total uncertainties for the collected η dataset.	48
4.1. Matched values collected for each flow rate parameter.	51

Nomenclature

ACR	=	advective capacity ratio, $\frac{\rho_c c_{p,c} U_c}{\rho_\infty c_{p,\infty} U_\infty}$
c_p	=	specific heat at constant pressure, (J/kg-K)
$C_p R$	=	specific heat ratio, $\frac{c_{p,c}}{c_{p,\infty}}$
D	=	cooling hole diameter, (m)
DR	=	density ratio, $\frac{\rho_c}{\rho_\infty}$
h	=	heat transfer coefficient, (W/m ² -K)
I	=	momentum flux ratio, $\frac{\rho_c U_c^2}{\rho_\infty U_\infty^2}$
l	=	arbitrary length scale, (m)
$\dot{m}$	=	flow rate, (SLPM)
M	=	mass flux (blowing) ratio, $\frac{\rho_c U_c}{\rho_\infty U_\infty}$
MW	=	molecular weight
P	=	pressure, (Pa)
PR	=	partial pressure
q''	=	heat flux, (W/m ²)
Re	=	Reynolds number, $\frac{\rho U l}{\mu}$
ReR	=	coolant-to-freestream Reynolds number ratio, $\frac{\rho_c U_c \mu_\infty}{\rho_\infty U_\infty \mu_c}$
s	=	entropy, (J)
T	=	temperature, (K)
T_{t_4}	=	combustor exit temperature, (K)
Tu	=	freestream turbulence intensity
U	=	velocity, (m/s)

VR	=	velocity ratio, $\frac{U_c}{U_\infty}$
x	=	first ordinate, (m)
y	=	second ordinate, (m)
δ^*	=	boundary layer thickness, (m)
ΔT	=	temperature difference, (K)
$\Delta \eta$	=	change in perceived adiabatic effectiveness
ε	=	measurement uncertainty
ζ	=	arbitrary function of measured quantities
η	=	adiabatic effectiveness, $\frac{T_\infty - T_{aw}}{T_\infty - T_c}$
$\bar{\eta}$	=	laterally-averaged adiabatic effectiveness
$\bar{\bar{\eta}}$	=	area-averaged adiabatic effectiveness
μ	=	dynamic viscosity, (Pa-s)
ξ	=	arbitrary measured quantity
π_c	=	compressor pressure ratio
ρ	=	density, (kg/m ³)

Subscripts

app	=	apparent
aw	=	adiabatic wall
c	=	coolant
D	=	based on hole diameter
s	=	surface
w	=	wall
$\emptyset$	=	no surface cooling
∞	=	freestream

Acronyms

CFD = computational fluid dynamics

IR = infrared thermography

PSP = pressure sensitive paint

SLPM = standard liters per minute

INFLUENCE OF COOLANT FLOW RATE PARAMETERS IN SCALING GAS TURBINE COOLING EFFECTIVENESS ON A FLAT PLATE

I. Introduction

Gas turbine engine design has evolved in many ways since the first engines were produced in the 1940s. Many of these changes were driven by the desire for more efficient and more powerful engines, as weight and performance are the two biggest parameters that influence aircraft design. All gas turbine engines operate on what is known as the Brayton cycle, which is seen plotted on a temperature-entropy, or T-s, basis in Figure 1.1. In order to increase the work output from this cycle, the area enclosed by the cycle lines must be expanded; this is commonly done in two different ways. The first is to increase the pressure rise, or π_c , from engine station 2 to 3 through the compressor. This results in raising the P_2 curve to a higher temperature at station 3, and if the same energy is added in the combustor, the total temperature at station 4 (T_{t_4}) will also increase. The second option is to keep π_c the same and simply increase the energy added in the combustor through burning fuel, again raising T_{t_4} . Both options push the station 4 point to expand the area under the cycle lines, resulting in more work done by the engine.

In most modern engines, the station 4 temperatures have been increased to values higher than the melting point of modern turbine blade materials. This results in the need to cool the turbine blades. The coolant flow is taken from the relatively cool compressor section via a bleed air valve and then piped through the turbine hub into the blades. Each blade is manufactured with open holes on the bottom to accept the coolant flow and distribute it throughout the blade. The process by which

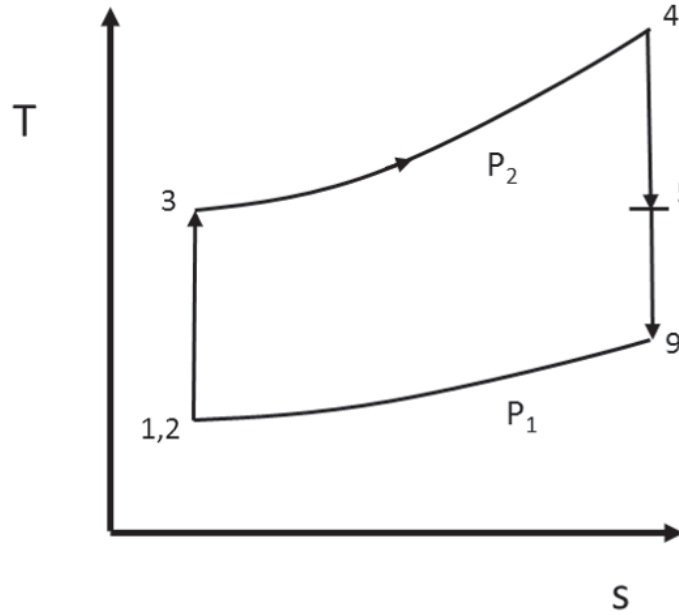


Figure 1.1. Brayton cycle with engine station numbers for a typical gas turbine engine.

energy is removed from the blade is categorized into two separate cooling methods: internal cooling and film cooling. Internal cooling is done by passing the coolant through small channels manufactured on the blade's interior. These channels are designed to maximize the exposed internal surface area to remove as much energy as possible. Film cooling is implemented by drilling holes through the blade's surface, and allowing the coolant to flow onto the exterior of the blade. This film of relatively cool fluid acts as a barrier between the hot freestream gases and the metal surface, thus reducing the heat flux into the blade.

The problem with both cooling schemes is that air taken from the compressor section is not able to participate in the combustion of fuel. Since energy was required to compress the air in the first place, taking oxidizer away from the reaction results in a loss of efficiency for the engine cycle. For this reason, turbine designers are constantly looking for new ways to efficiently use coolant so that the blades require less cooling flow to maintain structural integrity.

A large portion of the film cooling research focuses on how to optimize different hole shapes and layouts to efficiently cool a large area. Since testing new designs at engine operating temperatures in a lab can be costly and also complex, many engineers choose to test new designs with large, scaled-up models operating at room temperature conditions. The conditions of these tests are determined by matching various nondimensional values such as the freestream Reynolds number, the density ratio between the coolant and freestream gases, as well as the flow rate ratios between the coolant and freestream velocities. Engineers then take the low temperature, non-dimensional results and scale them up to predict cooling performance at the engine's operating conditions.

The scaling process is complicated by the non-linear nature of gas properties in high-temperature environments. Each gas property does not change in the same manner from high to low temperatures, making it difficult for researchers to sometimes match even one property ratio during low temperature testing. To quantify the coolant flow rate and other property ratios between the coolant and freestream gases, various coolant flow rate parameters have been developed. The main objective of this research is to determine the best coolant flow rate parameter to use during low temperature testing to scale the adiabatic effectiveness on a flat plate geometry.

At engine operating conditions, the density ratio between the coolant and freestream gases in the turbine section reaches values of approximately 2.0. This large relative change makes density the most affected gas property, but other gas properties are also affected. For example, the specific heat ratio between coolant and freestream gases is approximately 0.84. Due to this large density gradient, previous researchers have primarily used the density ratio to scale the adiabatic effectiveness. This has long been thought to yield the best results possible since the density ratio is the largest among the various property ratios. To test a density ratio other than 1.0, different

gases such as carbon dioxide (CO_2) and super chilled air have been used as coolant fluids. During room temperature testing, these gases typically provide for density ratios ranging from 1.2 to 2.0.

A good scaling ratio will correct for any changes in the density ratio, so using a gas with a wildly different density than air would truly test the ratio's scaling ability. A prime candidate for this is helium as it is one of the few extremely light gases that is also non-reactive. However, previous low temperature testing on a 90° leading edge injection rig exposed helium's tendency to create large temperature deviations on the surface, even when the coolant was at the same temperature as the freestream. Based on these results, a secondary objective of this research is to determine the suitability of helium to be used with the new flat plate geometry. If deemed suitable, helium is able to produce density ratios drastically lower than previously tested at around 0.15. Helium also possesses a quite different specific heat than air, a property that appears very little in scaling research. A perfect coolant flow rate parameter would still be able to scale the adiabatic effectiveness results when using gases with large property differences. The experiments in this research were conducted using infrared thermography (IR) as well as a variety of gases.

To summarize, the objectives of this study are:

1. Determine the best scaling parameter to use on a flat-plate, zero degree compound angle film cooling experiment using a variety of gases to generate variations in density and specific heat.
2. Determine if helium exhibits temperature separation effects during testing on a flat-plate, zero degree compound angle experiment.

II. Literature Review

Currently in the film cooling literature, most scaling research revolves around evaluating the proper scaling parameter to use during low temperature testing without having to match the density ratio. In order to provide the background literature and motivation behind the current study, this chapter will discuss various topic areas important to film cooling. These topic areas include the effects of hole shape and freestream conditions on film cooling (Section 2.1), the data collection methods (Section 2.2), and the process of scaling low temperature results up to engine operating conditions (Section 2.3).

2.1 Film Cooling Overview

Film cooling schemes are very common on the surface of turbine blades in gas turbine engines. Each scheme is implemented by creating holes in the surface of the blade, and then passing relatively cool fluid through the holes and onto the surface. If properly done, this will create a thin barrier of cool fluid separating the hot freestream from the part's surface, reducing the heat flux into the part. Bogard and Thole [3] outline many of the basic film cooling equations. They start by defining the heat flux at the test surface, seen in Equation 2.1, where h_f is the heat transfer coefficient with film cooling present, T_{aw} is the adiabatic wall temperature, and T_w is the measured wall temperature.

$$q_f'' = h_f(T_{aw} - T_w) \quad (2.1)$$

The adiabatic effectiveness, η , is a non-dimensional parameter defined using the adiabatic wall temperature, the coolant exit temperature, $T_{c,exit}$, and the freestream

temperature, T_∞ , as seen in Equation 2.2.

$$\eta = (T_\infty - T_{aw}) / (T_\infty - T_{c,exit}) \quad (2.2)$$

This is a common parameter used to quantify and compare different cooling hole geometries and layouts, as well as how the coolant spreads on the surface at different flow rates.

The flow rate of coolant is quantified using various ratios such as the blowing ratio (M), the momentum flux ratio (I), and the velocity ratio (VR). These are defined in Equations 2.3, 2.4, and 2.5 where ρ_c is the density of the coolant flow, ρ_∞ is the density of the freestream flow, U_c is the velocity of the coolant flow, and U_∞ is the velocity of the freestream flow. These parameters are also used when scaling lab results up to engine operating conditions. Past research has shown that each of these parameters scale η well under specific operationg conditions, but none have been found to be a universal scaling parameter. Scaling will be discussed further in Section 2.3.

$$M = \rho_c U_c / \rho_\infty U_\infty \quad (2.3)$$

$$I = \rho_c U_c^2 / \rho_\infty U_\infty^2 \quad (2.4)$$

$$VR = U_c / U_\infty \quad (2.5)$$

2.1.1 Hole Shape Effects

Hole shape is important in film cooling research as different geometries spread the coolant on the surface differently. The simplest geometry, the cylindrical hole, is very susceptible to jet lift-off at higher blowing ratios. This was the main focus of a study done by Baldauf et al. [2]. It is seen in Figure 2.1 that as blowing ratio was increased, the point of maximum adiabatic effectiveness moved downstream and the

magnitude of the maximum η value decreased. This was due to the cooling jet lifting off the surface close to the hole exit and then reattaching further downstream. The longer the coolant flow stays detached above the surface, the more the coolant fluid heats up, causing the drop in magnitude. Because of this jet lift-off, cylindrical holes are limited to lower blowing ratios to remain efficient in spreading cool fluid onto the surface of the part. This is not desirable in engine design as the amount of coolant flowing through the holes cannot always be controlled.

A desire for better hole geometries gave birth to two new designs: the fan-shaped hole and the laid-back fan-shaped hole. These designs essentially widened the exit of the cooling hole which slows down the coolant flow, reducing the tendency for separation from the part's surface. Gritsch et al. [7] were the first to study the differences between cylindrical, fan-shaped, and laid-back fan-shaped cooling holes. The differences in the geometries can be seen in Figure 2.2. The addition of a fan-shape and laid-back angle, provided increased η values in both the lateral and streamwise direction, seen in Figure 2.3. Furthermore, with an increase in blowing ratio, the η values for the shaped holes increased, most notably on the laid-back hole. This validated the new designs as they spread the coolant more efficiently onto the surface and reduced the risk of wasting coolant through jet lift-off.

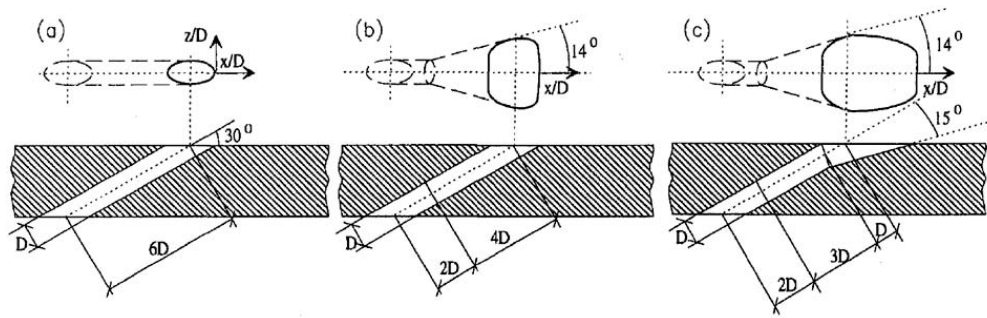


Figure 2.2. Three hole geometries: a) Cylindrical b) Fan-Shaped c) Laid-Back Fan-Shaped [7].

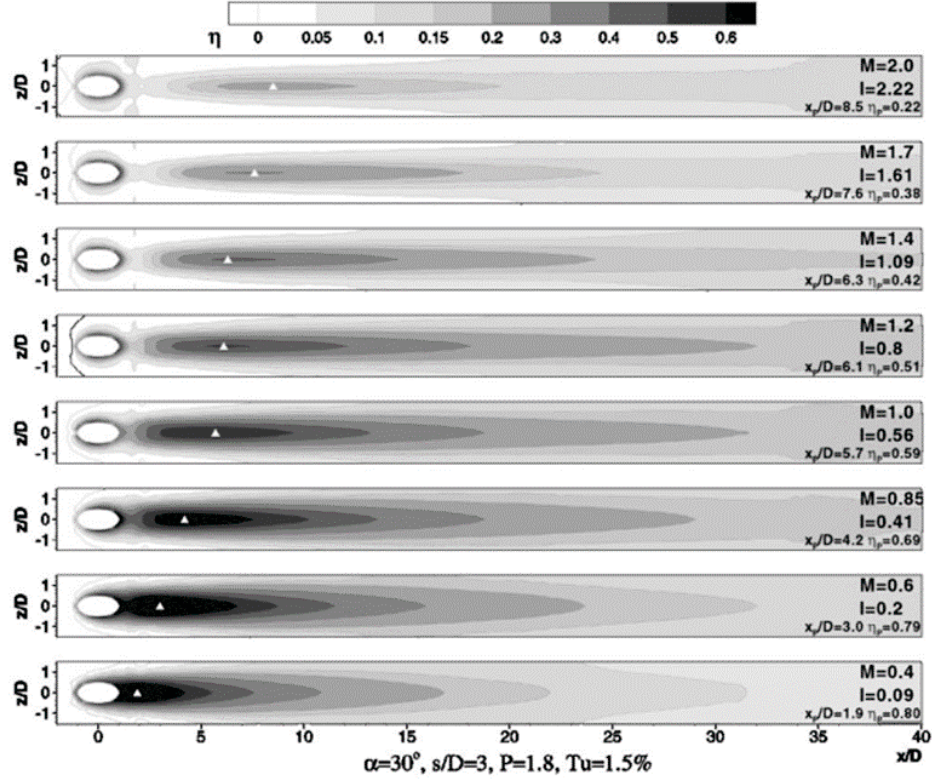


Figure 2.1. Adiabatic effectiveness values for a cylindrical hole at various blowing ratios [2].

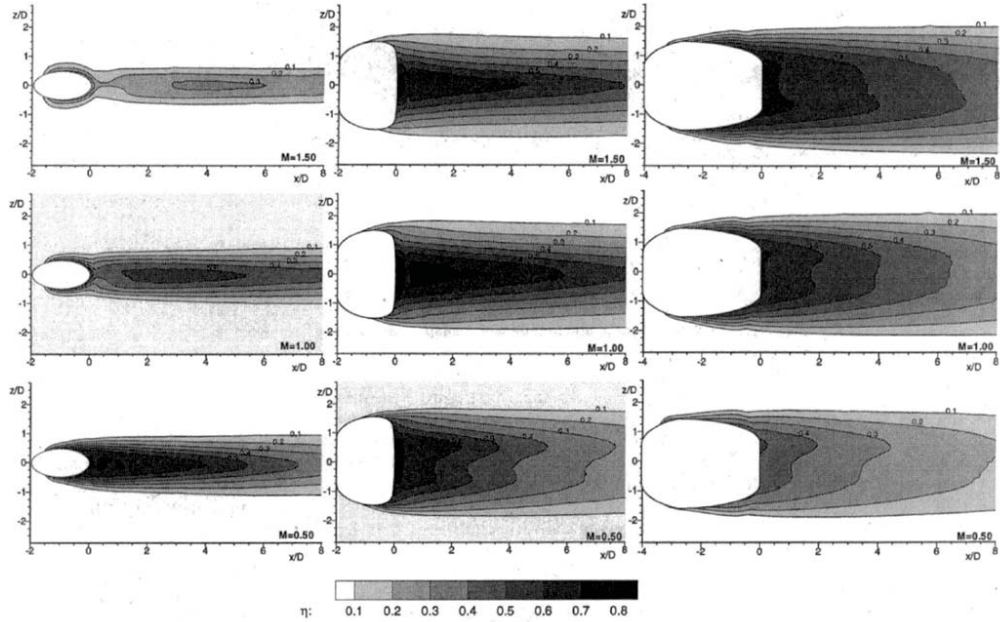


Figure 2.3. Adiabatic effectiveness distributions for the three hole geometries pictured in in Figure 2.2 [7].

The increase in cooling efficiency is also seen when the shaped holes are used in full coverage layouts. This was the focus of a study conducted by Dittmar et al. [4]. The authors studied the differences between different cooling hole layouts using cylindrical holes, slots, and fan-shaped holes, pictured in Figure 2.4. Again they found that the fan-shaped holes performed better compared to both the cylindrical holes and the slots in the lateral and streamwise directions. Figure 2.5 shows these results for both low and high blowing ratios. These results occurred because the fan-shaped holes diffuse and spread the coolant evenly onto the surface, which also prevents the coolant jet from lifting off at higher blowing ratios.

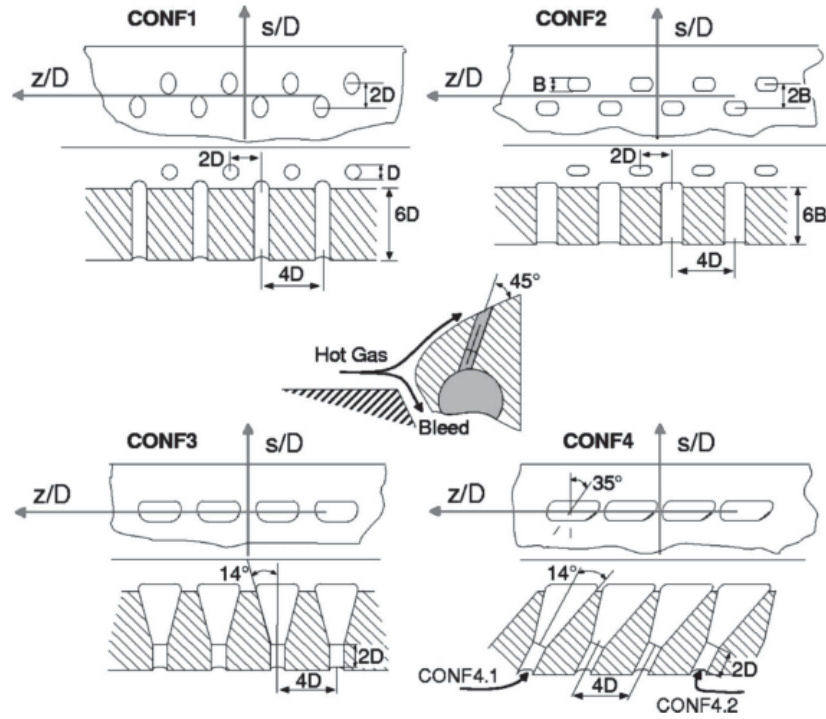


Figure 2.4. Four different cooling hole layouts used by Dittmar et al. [4].

Since fan-shaped and laid-back fan-shaped holes have many geometric parameters that can be adjusted and optimized, many of the developed shaped hole geometries are proprietary. To combat this, Schroeder and Thole [13] developed a baseline academic geometry that is now available for researchers to use in shaped hole research. To

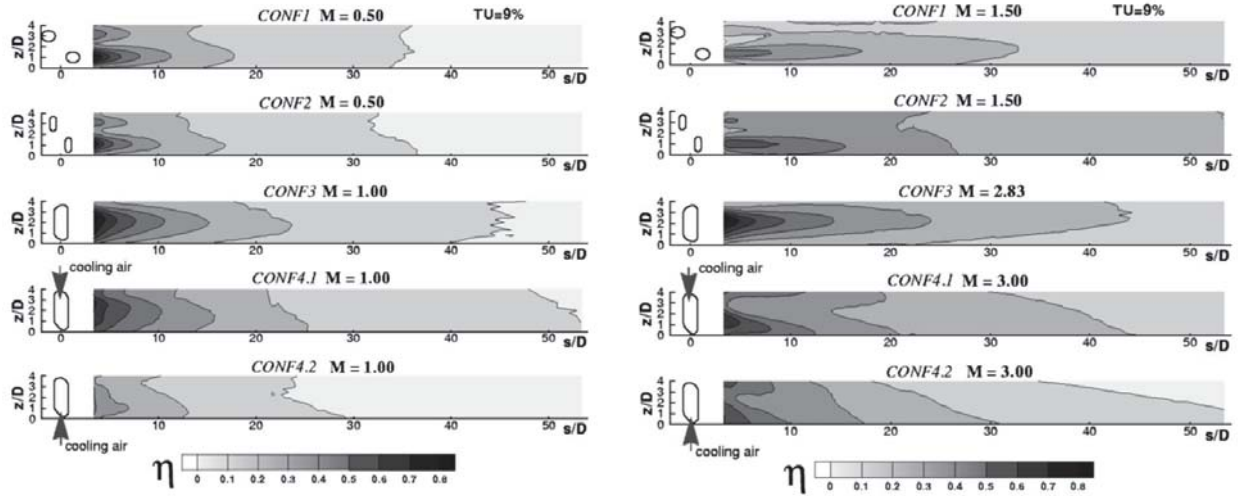


Figure 2.5. Adiabatic effectiveness for five different test configurations at a low blowing ratio [4].

develop this geometry, the authors studied many different hole shapes presented in the open literature and chose parameters that fell approximately in the middle of the ranges, as seen in the table of Figure 2.6. This resulted in the 'triple 7', or 7-7-7, hole. It is named as such because the hole exit expands seven degrees in the positive and negative y-direction, and includes a seven degree laid-back angle, as seen in the layout of Figure 2.6. Now, shaped hole research can be conducted using this baseline geometry and researchers can accurately compare results with each other.

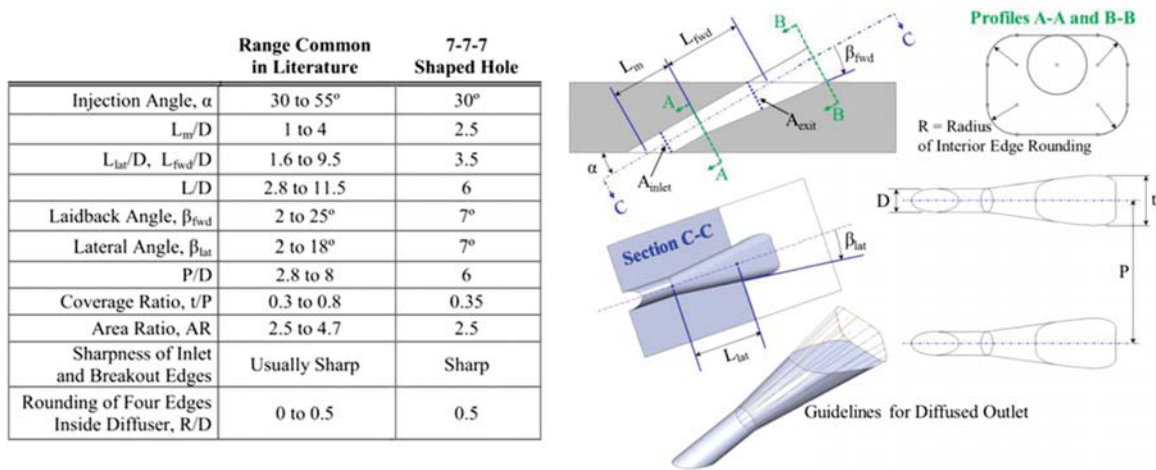


Figure 2.6. 7-7-7 hole design parameters and layout [13].

2.1.2 External Flow Effects

In addition to the cooling hole shape, the state of the external flow around the hole and coolant plume also affects how coolant is spread onto a part's surface. The factors that best describe the film cooling environment are freestream Mach number, Reynolds number, and boundary layer thickness. Anderson et al. [1] studied the effects that each of these factors had on the η distributions for shaped holes. In this study, the authors used the 7-7-7 shaped hole and attempted to independently vary each factor using a variety of geometries and flow conditions. For the Mach number, they found that over a range of 0.08-0.15, laterally averaged η values varied by a maximum of 0.04, which was within the experimental uncertainty. The authors attributed this variation to the inability to control the boundary layer thickness, as δ^*/D varied from 0.08-0.29. After analyzing η profiles at two downstream locations, the authors found that the values from three tests were all within experimental uncertainty, and thus stated that Mach number has a negligible effect on η .

To show the effect of boundary layer thickness, the authors varied δ^*/D from 0.11 to 0.60 while holding the Reynolds number (Re_D) constant at 5,500. They found that the thickest boundary layers resulted in the highest values of η , especially with high freestream turbulence. They attributed this increase in performance to increased turbulence in the thick boundary layer, which pushed a slightly separated jet closer to the surface.

The final part of the study done by Anderson et al. [1] analyzed the effect that Reynolds number had on η . In the two test cases of $Re = 15,000$ and $5,500$, the boundary layer thickness (δ^*/D) varied from 0.08 to 0.11. They discovered that at lower blowing ratios, an increase in the Reynolds number yielded an increase in the laterally averaged η values, as shown in Figure 2.7.

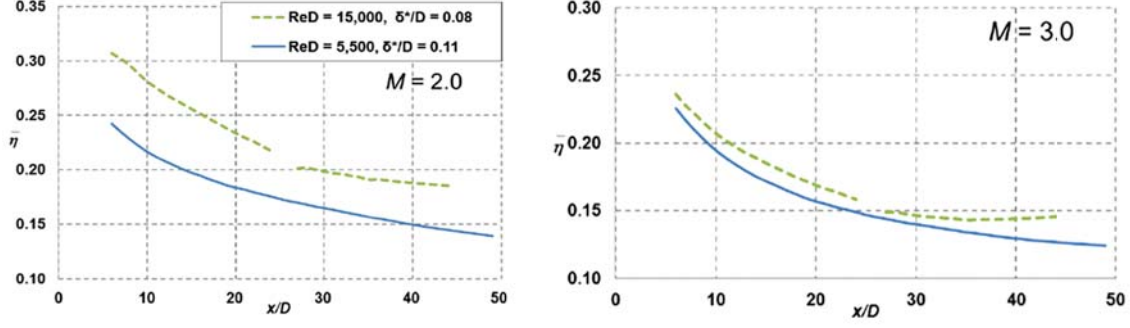


Figure 2.7. Effect of Reynolds number on laterally-averaged η [1].

2.2 Measurement Techniques

There are many methods used to collect data when conducting film cooling experiments. The ultimate goal of each method is to accurately quantify the effects of the coolant plume on the test article without disrupting the cooling or freestream flow. The methods can be categorized into thermal methods and mass-transfer methods.

2.2.1 Thermal Methods

Thermal measurement techniques are designed to directly observe the surface temperature during the experiment. Two of the most common thermal methods are infrared thermography (IR) and the temperature sensitive liquid crystal method. The IR method is much more common in modern literature as it does not require a special surface treatment and the only component required is the IR camera. Once the surface temperatures are obtained using the desired method, the coolant and freestream temperatures are directly measured by thermocouples. These three values are then used to calculate an apparent η (η_{app} , Equation 2.6), which requires a conduction correction in order to produce the true η value.

$$\eta_{app} = (T_{\infty} - T_s) / (T_{\infty} - T_{c,exit}) \quad (2.6)$$

A conduction correction is required for any thermal measurement during η experiments because no surface is truly adiabatic. Any conduction through the test article will lower the observed surface temperature and thus affect the resulting η_{app} distribution. To correct η_{app} for conduction, Williams et al. [18] used a simple formula, seen in Equation 2.7.

$$\eta = \frac{\eta_{app} - \eta_0}{1 - \eta_0} \quad (2.7)$$

In this equation, η_{app} is the nondimensional surface temperature collected during the test run while η_0 is the nondimensional surface temperature with the same internal cooling flow, but with no coolant flowing on the external surface. Therefore, η_0 represents a sort of 'bias' in η due to conduction through the test article. This method assumes 1-D conduction and can be used on most low conductivity models.

Klavetter et al. [9] took this conduction correction one step further by use of a 3-D conduction analysis code called Comsol Multiphysics[®]. Using the measured temperatures as boundary conditions, the code calculated the expected η_0 seen on the test surface. By comparing these results to the expected η_0 values from a 1-D assumption, the authors found that the results were very similar in nature, only varying by ± 0.05 . The distribution of the difference between the two calculations is seen in Figure 2.8. As expected, the highest differences exist in the near-hole region. This can be attributed to the material being thin near the hole cutouts, as well as high temperature gradients which allow for conduction laterally.

As mentioned previously, the IR method is widely used in film cooling research due to its ease and simplicity over other methods such as liquid crystals. The method employs the use of an IR camera which captures the surface temperature of the test article. This is accomplished via calibrating the emitted thermal radiation to the surface temperature. To calibrate the camera, Baldauf et al. [2] used a technique where thermocouples were attached flush with the test surface. IR images were then

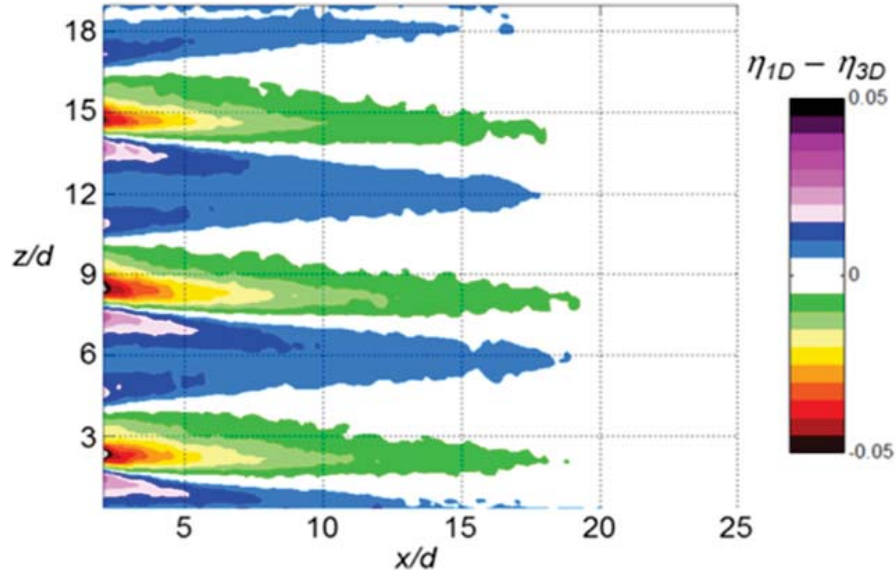


Figure 2.8. Difference in η_0 values calculated with 1D and 3D conduction assumptions [9].

taken at a steady state condition and the thermocouple locations were then located on the IR images. A curve-fit was then applied to the distribution of data points and a calibration curve was generated between the collected IR counts and thermocouple temperature readings. This technique is quite versatile, as it accounts for the radiation effects from the tunnel walls as well as the effects of the viewing window's transmission and emission, thus providing an 'all-in-one' calibration for the specific rig and tunnel configuration.

The issue with thermal methods is that surface boundary conditions such as an adiabatic wall are impossible to replicate as no material is truly adiabatic. The desire to directly test these boundary conditions without the use of computational fluid dynamics (CFD) led to the development of mass-transfer methods.

2.2.2 Mass-Transfer Methods

Mass-transfer methods were developed using the mass-transfer analogy, which equates the transfer of mass through a system to the transfer of heat through the same system. These methods are common in the film cooling literature as they allow for the exact implementation of wall boundary conditions such as adiabatic walls.

One of the most common mass-transfer methods for implementing the adiabatic wall boundary condition is the use of Pressure Sensitive Paint (PSP). In order to observe the transfer of mass through the system, a foreign gas is used as the coolant flow and the fraction of oxygen present at the surface is measured by way of partial pressures. PSP is used as this indicator of the partial pressure of oxygen present at the surface. As seen in Figure 2.9, there are two types of PSP: single component and binary. Single component PSP was the first to be developed, which included only a pressure-sensitive luminescent molecule in the polymer binder that emits at a specific wavelength when oxygen is present. The problem with this paint was that the pressure-sensitive luminophore was also temperature sensitive, so if the coolant temperature was not exactly the same as the freestream temperature, the measured partial pressure of oxygen would not be accurate and would need to be corrected. This temperature sensitivity led to the advent of binary PSP. Binary PSP includes a second luminophore which is temperature-sensitive, but not pressure-sensitive. The second luminophore emits at a different wavelength than the pressure-sensitive luminophore, so by measuring the intensity at both wavelengths, the partial pressure of oxygen can be correlated to the ratio of measured intensity to a reference intensity. The partial pressure of oxygen on the surface can then be converted into an η measurement by means of Equation 2.8.

$$\eta = 1 - \left[1 + MW \left(\frac{PR_{O2,air}}{PR_{O2,surf}} - 1 \right) \right]^{-1} \quad (2.8)$$

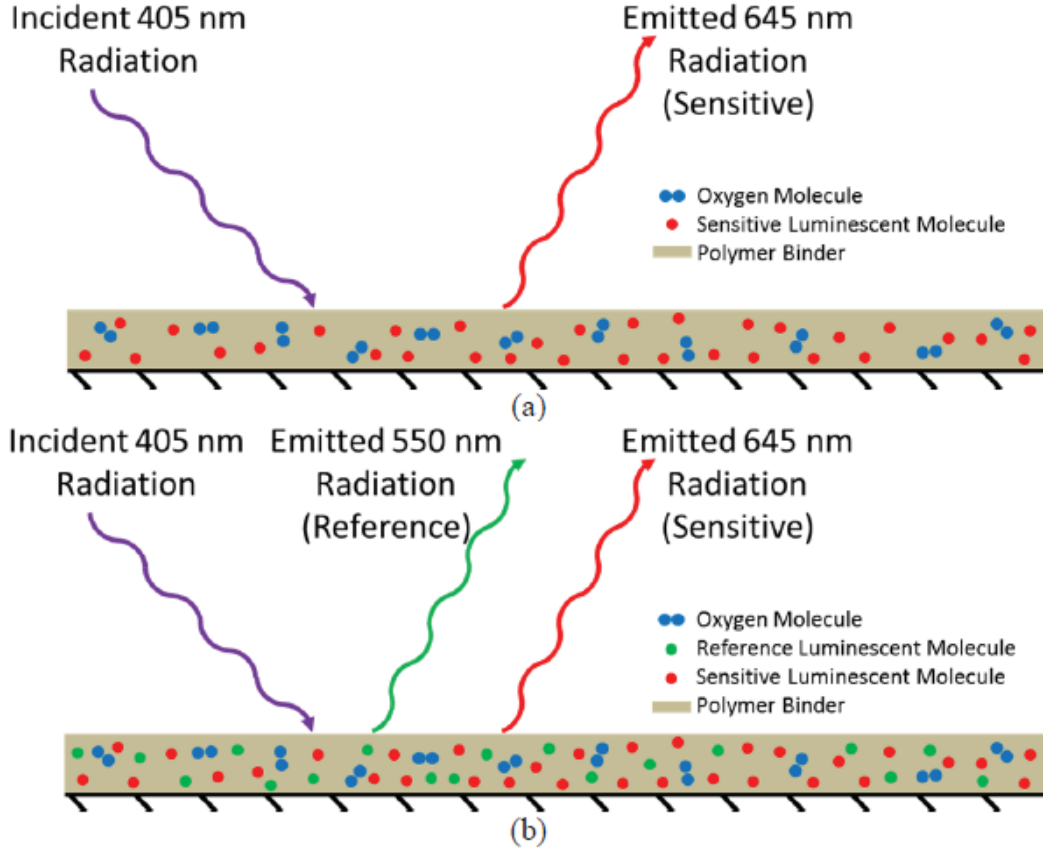


Figure 2.9. Structure of both a) single-component PSP and b) binary PSP [15].

The problem with using mass-transfer methods is that film cooling relies on both mass and thermal diffusion. By ignoring the thermal diffusion, the methods do not account for increasing coolant gas temperatures, resulting in an over-prediction of η . Wiese [15] specifically analyzed the differences between the thermal and mass-transfer methods, and documented this over-prediction. As seen in Figure 2.10, when the same coolant flow was measured with thermal and PSP methods, the maximum η values from PSP were consistently higher than the thermal values, and the PSP η_{max} profiles decayed at a lower rate. The thermal method accounts for not only coolant molecules diffusing into the freestream, but also the same coolant molecules gaining energy through thermal diffusion, thus resulting in a steeper η_{max} decay.

Due to the inability of PSP methods to capture thermal diffusion, the IR method

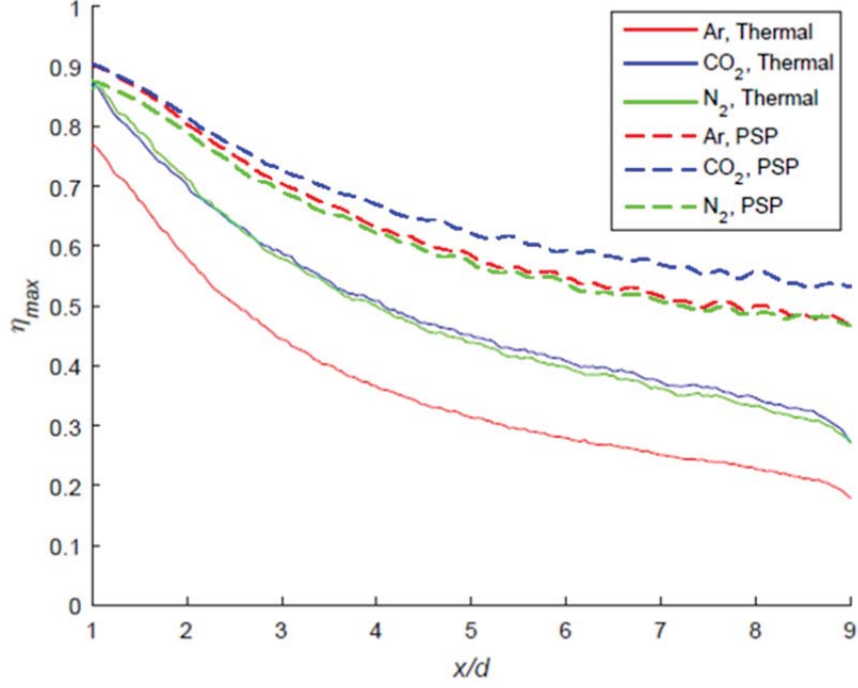


Figure 2.10. Resulting η_{max} values at all x/d locations from thermal and PSP measurements ($I = 1.0$) [15].

was selected as the appropriate measurement technique because the current study is interested in determining which properties should be accounted for while scaling η values. Since thermal methods are able to capture the effects of both mass and thermal diffusion, the IR method will provide the best evidence to support the final conclusions.

2.3 Scaling

Scaling is the process by which gas turbine component temperatures are predicted using the results obtained in a low temperature lab setting. This would initially seem quite simple as the obtained η values are nondimensional, so the wall temperatures could be predicted by plugging in the expected freestream and coolant temperatures. However, this is not the case due to the non-linear nature of fluid properties with increasing temperature. As seen in Table 2.1, the properties of the various gases used

in low temperature testing do not vary in the same manner as at engine conditions. For this reason, much of the scaling research has focused on identifying coolant flow rate parameters that are not influenced by multiple changing fluid properties.

Table 2.1. Properties of various gases for engine and testing conditions [8].

Gas	Density, ρ (kg/m ³)	Specific Heat, c_p (kJ/kg-K)	Dynamic Viscosity, μ 10 ⁻⁶ (Pa-s)	Thermal Conductivity, k 10 ⁻³ (W/m-K)
Engine Freestream (Air, 2000 K, 30 bar)	5.22	1.34	68.9	137
Engine Coolant (Air, 1000 K, 30 bar)	10.45	1.14	42.44	66.7
Air (300 K)	1.16	1.01	18.46	26.3
Cold Air (150 K)	2.34	1.01	10.34	13.8
CO ₂	1.77	0.85	14.9	16.55
N ₂	1.12	1.04	17.82	25.9
Ar	1.66	0.52	22.3	17.2
He	0.16	5.19	19.9	152

2.3.1 Density Effects

Much of the previous scaling research has focused on the effects of the density ratio (DR), or the ratio of the coolant and freestream densities, seen in Equation 2.9.

$$DR = \frac{\rho_c}{\rho_\infty} \quad (2.9)$$

The thought was that if the appropriate scaling parameter was used, varying the DR would have little to no effect on the η profiles and would closely predict the component's adiabatic wall temperatures. This was the premise behind a study done by Sinha et al. [14]. To test the theory, the researchers used a simple flat plate geometry with axially-oriented cylindrical cooling holes. They quantified the coolant flow rate by the three previously defined parameters, I , M , and VR , and then varied

DR . The DR changes were achieved by drastically reducing the temperature of the air to be used as coolant. At its extreme, the coolant was chilled to 150K, while the freestream was held constant at 300K to produce a $DR = 2.0$, as seen in Table 2.1. The challenge was to figure out which flow parameter best predicted the η distributions. If the measured values of η did not change between different values of DR at the same flow rate parameter, then that particular flow rate parameter best predicted the actual η distribution. As seen in Figure 2.11, none of these parameters perfectly scale η , but the authors claim that I predicts η best with a max deviation of 0.15.

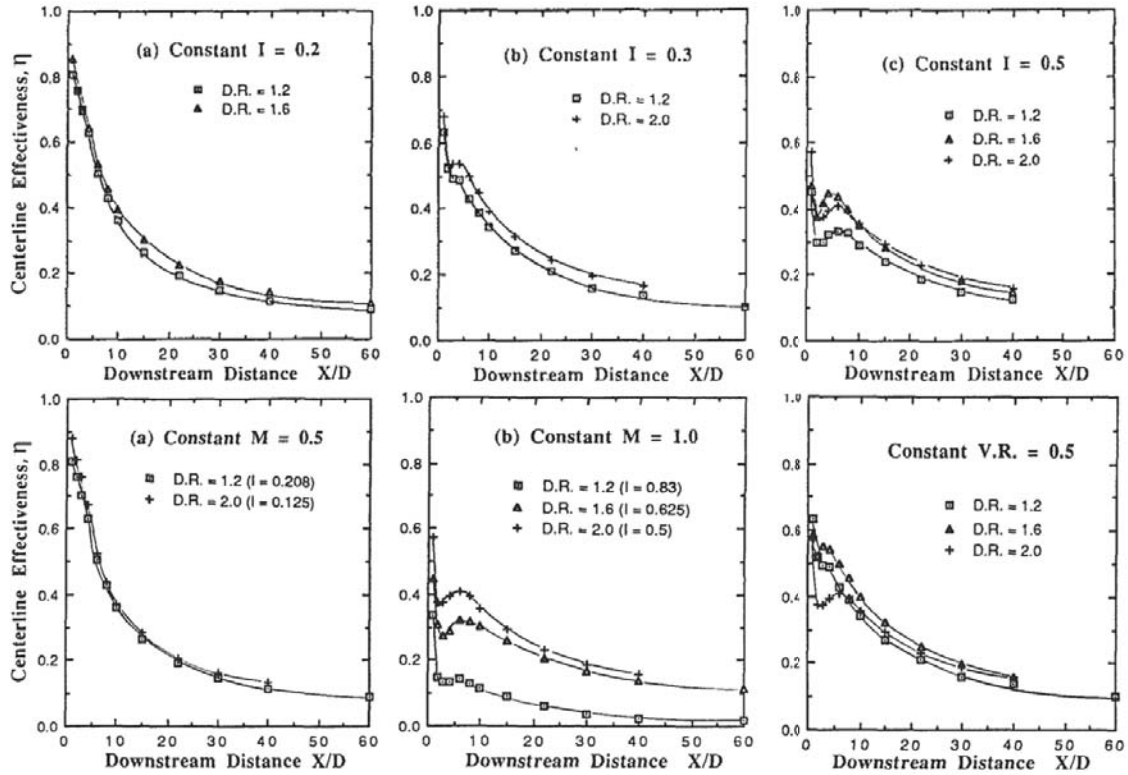


Figure 2.11. Matched I , M , and VR cases with varying DR . Adapted from [14].

This was compared to the matched M experiments which had a max deviation of 0.25, and the matched VR cases which had a max deviation of 0.20. However, it should be noted that the researchers only matched values up to 0.5 of I , but when M

was matched at 1.0, the highest tested I became 0.83 in the $DR = 1.2$ case. As seen before in Figure 2.1, cylindrical holes are very prone to jet separation, so it is very possible that the large change in η for the matched $M = 1.0$ case could be caused by jet separation. This separation was not seen in the matched I cases because the matched values were less than 0.5, resulting in somewhat inconclusive results as to which parameter best scales η .

The theme of inconclusive results surrounding variable density ratio research continued in a study done by Ethridge et al. [6]. In this study, the researchers were examining the centerline adiabatic effectiveness of a cylindrical cooling hole located on the suction side of a turbine blade. By matching both M and I , the researchers again sought to determine the best scaling parameter for η . As with the previous study, the DR was varied using low temperature air as coolant. As seen in Figure 2.12, neither M nor I scaled η between the two density ratios.

The authors claim that M does best when $M < 0.7$, but with an exception at $DR = 1.1$ and $M < 0.3$, where the performance is quite different. At higher M values, specifically $I > 0.4$, the authors say that I scales best between density ratios. The authors do claim that these results are consistent with the expected outcomes as,

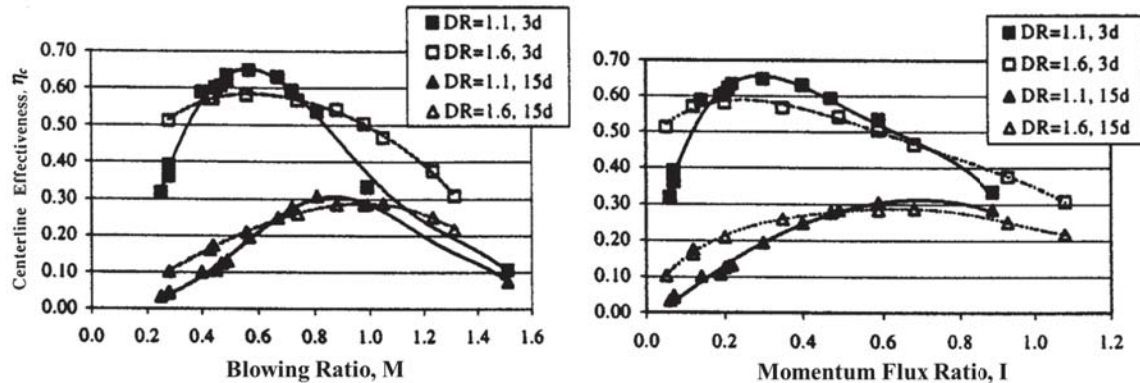


Figure 2.12. η distributions at matched M and I with varying DR . Distributions from $x/D = 3$ and 15 are plotted. Adapted from [6].

“better scaling with M should occur at blowing ratios low enough that the coolant jets remain attached to the surface, while better scaling with I should occur when the jet begins to detach” [6]. Even though they were consistent with previously published literature, the results had to include exceptions as neither parameter did a perfect job.

Another method of changing the density ratio is to use a coolant gas with a higher density than air, such as CO_2 . Ekkad et al. [5] utilized this method to evaluate the scaling abilities of M . The study used a cylindrical hole on a flat plate with a 0° injection angle. As seen in Figure 2.13, the resulting laterally-averaged η values ($\bar{\eta}$) when using CO_2 as the coolant gas were most often higher than when air was the coolant gas. The authors attribute this to the jet’s momentum as, “the CO_2 jet has a lower momentum, which could cause the coolant to stay closer and protect the surface better” [5]. This statement is consistent with prior research on cylindrical holes, however the authors do not conduct tests matching I which would show the effects of jet momentum.

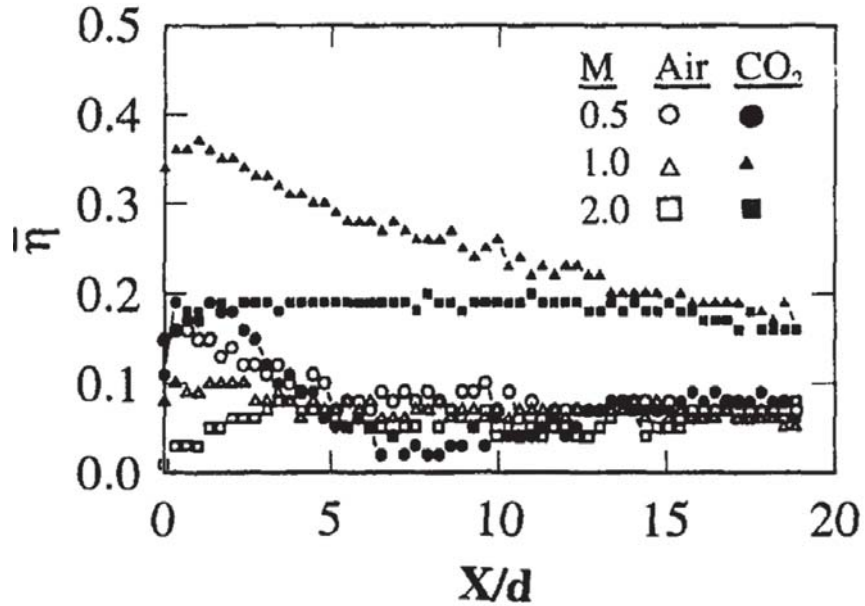


Figure 2.13. $\bar{\eta}$ distributions from three matched M values using air and CO_2 as coolant. Adapted from [5].

The scaling problem is complicated by external flow effects as a result of differing model geometries. As described in Section 2.1.1, the shape of the cooling hole drastically changes the placement of coolant flow on the surface. This change in coolant placement could change which parameter best scales lab results. This was evident when Schroeder and Thole began testing the 7-7-7 hole [13]. Through their testing, they found that, “effectiveness for the shaped holes did not scale with blowing ratio or momentum flux ratio over the range evaluated” [13]. Figure 2.14 shows the mismatched area-averaged η values at the two tested density ratios. The DR was varied using liquid-nitrogen chilled air as the coolant gas. These results show that although the density ratio does influence scaling performance, it is not the only factor to consider when attempting to scale adiabatic effectiveness results.

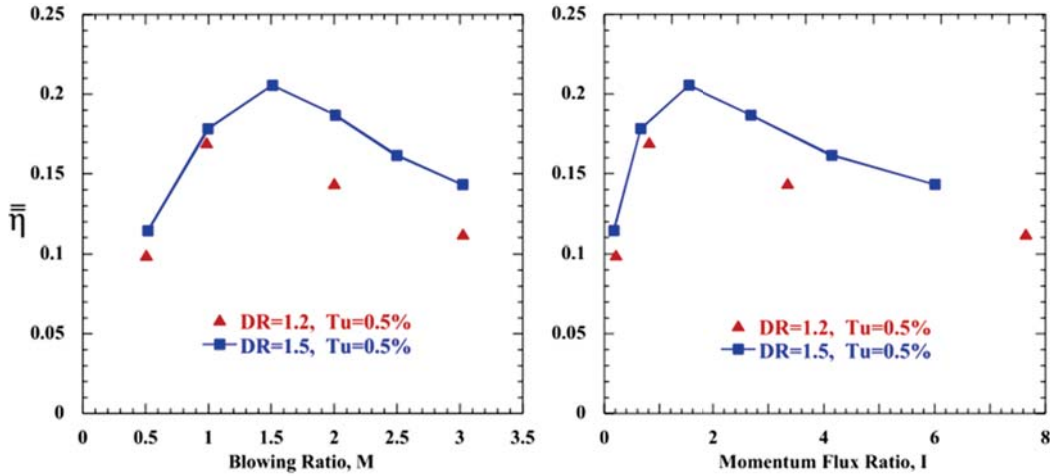


Figure 2.14. Area-averaged η at various M and I values. Averaged over $x/D = 3-35$. Adapted from [13].

2.3.2 Temperature Separation Effects

In an attempt to significantly alter the density ratio, with DR values less than 1.0, Wiese [15] tried to use helium (He) as a coolant gas. The geometry used was a semi-circle leading edge, with a cylindrical coolant hole placed 21.5° off the stagnation point. The hole was drilled perpendicular to the freestream direction, so the coolant

was injected at a 90° compound angle. During the He testing, it was noticed that under certain coolant flow conditions, the measured temperature on the surface was hotter than the freestream temperature. To investigate, the coolant and freestream temperatures were set to the same value, so the entire surface was expected to be at the same temperature. A sweep of matched I values was then conducted to capture any discrepancies. The largest differences were seen at the $I = 0.25$ case, seen in Figure 2.15. Even though the coolant and freestream temperature were within 0.1K, the measured surface temperatures were $\pm 10\text{K}$ with helium, but less than 1K of difference when using other gases. Wiese was unable to determine the cause of this “energy separation phenomenon”, but does offer a variety of hypotheses ranging from the fluid acting like a Ranque-Hilsch tube to kinetic energy effects of monatomic and diatomic gases [15]. If the presence of this “energy separation phenomenon” had not been detected during testing, then helium would have been a viable option for Wiese to drastically alter the density ratio, thus potentially leading to more conclusive results on which parameter best scales adiabatic effectiveness.

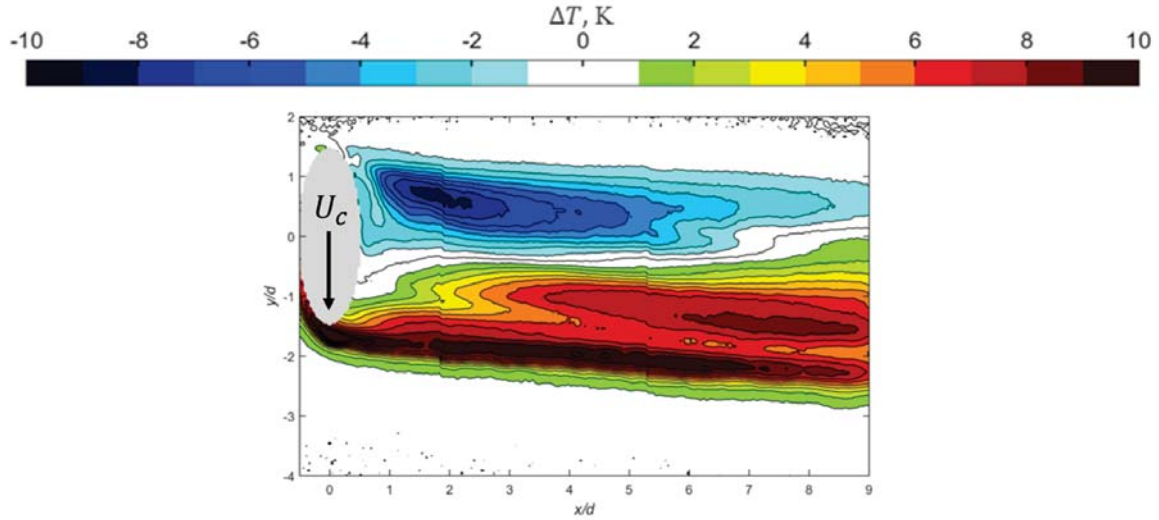


Figure 2.15. ΔT distribution for He at $I = 0.25$, ($T_\infty - T_c = 0.1\text{K}$). Adapted from [15].

2.3.3 Specific Heat Effects

The wide-spread theme of inconsistent results from previous density ratio studies led researchers to seek out other parameters that had an influence other than just the density and velocity of the coolant and freestream flows as accounted for in M , I , and VR . This was the motivation behind a study done by Rutledge and Polanka [12]. In addition to M , I , and VR , the authors also looked at the effects of matching the Reynolds Number ratio (ReR), as well a new parameter defined by the authors: the heat capacity ratio, or as it is now known, the Advective Capacity Ratio (ACR), seen in Equation 2.12. This parameter includes the specific heat ratio, or C_pR , between the coolant and the freestream gases.

$$ReR = \frac{\mu_\infty \rho_c U_c}{\mu_c \rho_\infty U_\infty} \quad (2.10)$$

$$C_pR = \frac{c_{p,c}}{c_{p,\infty}} \quad (2.11)$$

$$ACR = \frac{c_{p,c} \rho_c U_c}{c_{p,\infty} \rho_\infty U_\infty} \quad (2.12)$$

This parameter was developed due to the suspected impact that c_p has on η distributions. To test this, the authors simulated a turbine blade leading edge in CFD and then used various coolant flow rates with air and a foreign gas, CO_2 . The baseline case was run with air as a coolant, and a temperature of 20K less than the freestream. The properties of the coolant were then artificially adjusted to fix the density ratio at a value of 1.0, and the flow rate was adjusted so that all ratios, M , I , VR , ReR , and ACR , were also set at a value of 1.0. Then they switched the coolant gas to CO_2 at the same temperature which resulted in a density ratio of approximately 1.5. Through five separate cases, the flow rate was adjusted to set M , I , VR , ReR , and ACR to a value of 1.0 individually [12]. Figure 2.16 shows the resulting η distribu-

tions at an x/D location of 3. As one can see, the matched *ACR* case closely matches the baseline distribution in terms of both magnitude and also location on the surface, even though the CO_2 case was at a density ratio of 1.5. The authors do point out that the *M* case best matched the maximum magnitude of η and *I* best predicted the peak location. However, *ACR* proved to be a good compromise between *M* and *I*, thus they recommend testing *ACR* in future laboratory studies.

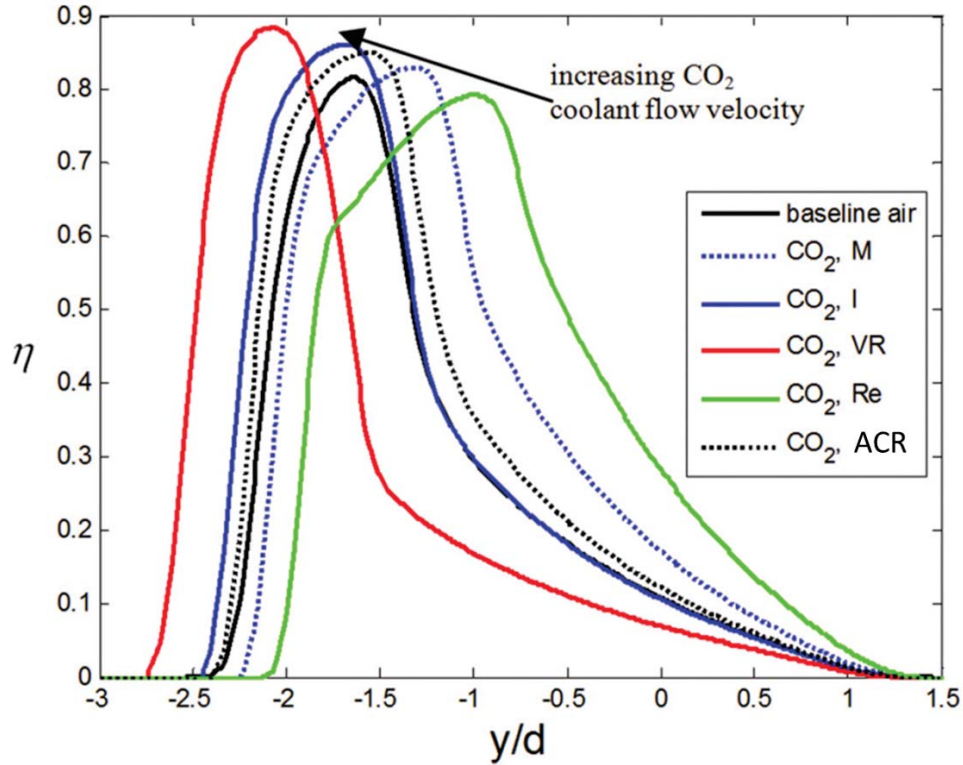


Figure 2.16. η distributions matching various coolant flow rate parameters [12].

Motivated by the success of *ACR* in the CFD study, Wiese et al. [16] set out to test *ACR* on a leading edge model similar to the one tested in the previous CFD study. The authors used the same open loop tunnel used for the current research and quantified the results using thermal IR techniques. They ran multiple foreign gases including Argon (Ar), CO_2 , and N_2 as well as air for coolant flow to induce both density ratio and specific heat ratio changes. By matching each of the five flow

rate parameters, M , I , VR , ReR , and ACR , the researchers attempted to quantify which parameter would scale η best. As seen in Figure 2.17, the results showed that when I was matched, the peak values of η occurred at the same y/D location, but the magnitude of η varied among the gases. This prediction ability of I was not seen with the other traditional scaling parameters, M and VR . Neither M or VR predicted the coolant plume location or magnitude of η between all gases as seen in Figure 2.18.

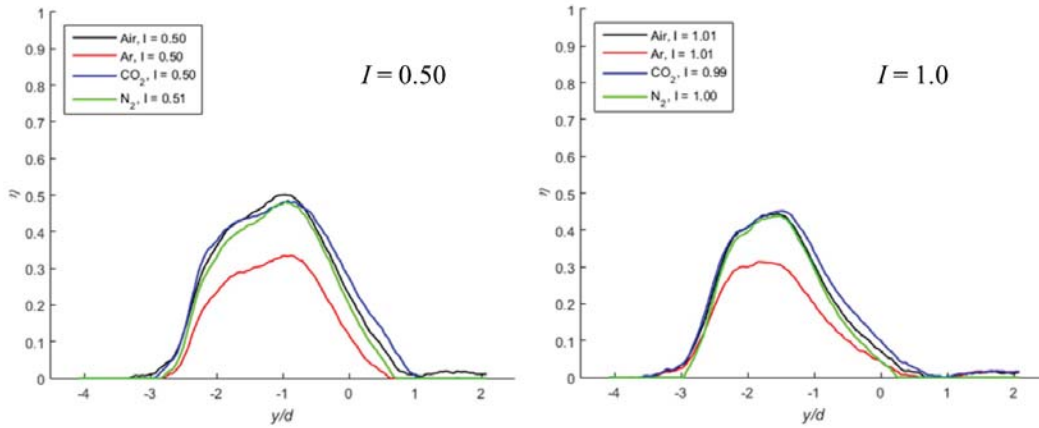


Figure 2.17. η distributions matching $I = 0.5$ and 1.0 at $x/d = 5.0$ [16].

Wiese also presented matched ACR cases in his thesis [15] using both IR and PSP methods. As seen in Figure 2.19, when ACR was matched, the η distributions of air, CO_2 , and N_2 closely matched in both magnitude and distribution shape. However, the Ar profile was vastly different and produced very low η values in the $ACR = 2.0$ case. Wiese attributes this to jet separation effects. Since the specific heat of Ar is significantly lower than the other gases used, matching ACR leads to very high values of I . Since I has been established to be a good indicator of jet separation especially with cylindrical holes, high values would indicate a separated jet. In the case of $ACR = 2.0$, the I value associated with Ar was approximately 10.1 [15]. This would explain the extremely low η profile for Ar in Figure 2.19. This figure also reiterates the over-prediction of η that occurs when PSP methods are used, discussed in Section 2.2.1.

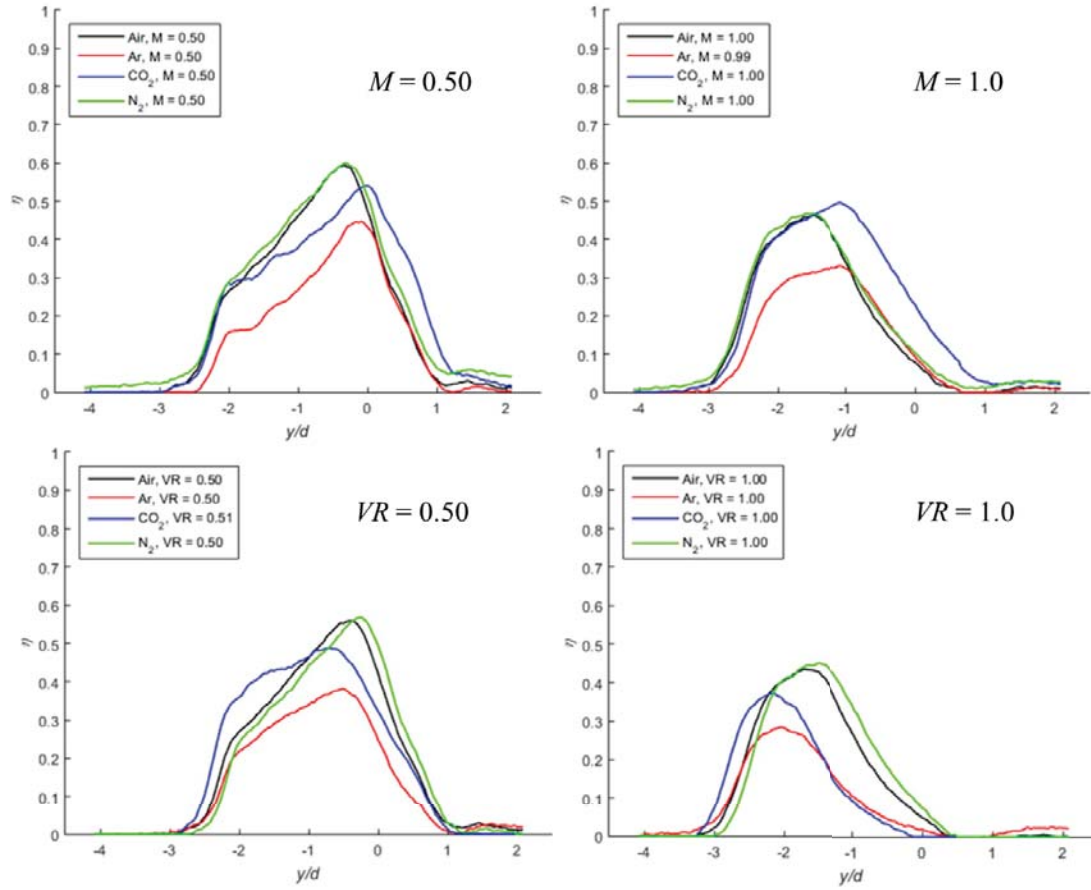


Figure 2.18. η distributions matching M and VR at 0.5 and 1.0 at the location $x/d = 5$. Adapted from [16].

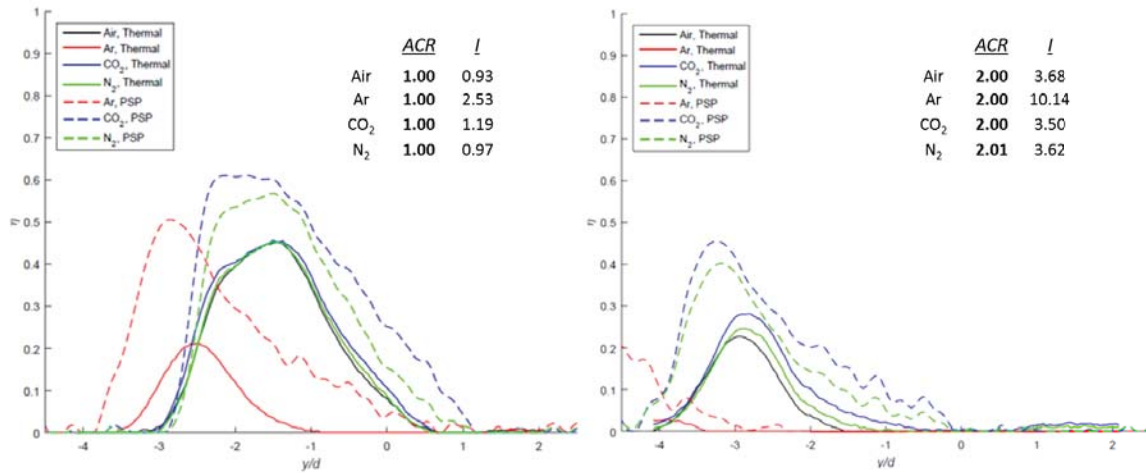


Figure 2.19. η distributions at $x/d = 5$, matching ACR using IR and PSP methods. Adapted from [15].

Another difference seen between the two ACR cases is that the CO_2 , N_2 , and air profiles do not match as well at $ACR = 2.0$. Wiese examines this using Figure 2.20, where a full ACR sweep is plotted for both air and CO_2 at the same x/D location. This clearly shows how the coolant plume location and effectiveness change with changing flow rate. One can see that although ACR does a decent job at matching the plume location between gases, the magnitude of each profile is different. Wiese attributes the differences to both separation effects in terms of high I values at high ACR values, and also viscosity differences between the two gases which would change how the gases mix with the freestream flow. Wiese concludes his ACR discussion by suggesting that further research must be done, “to fully characterize the merits of ACR on scaling adiabatic effectiveness magnitude. Matched ACR experiments should be conducted at flow conditions that are not momentum dominated, such as on a flat plate with zero compound angle injection” [15], which is exactly what the current study aims to do.

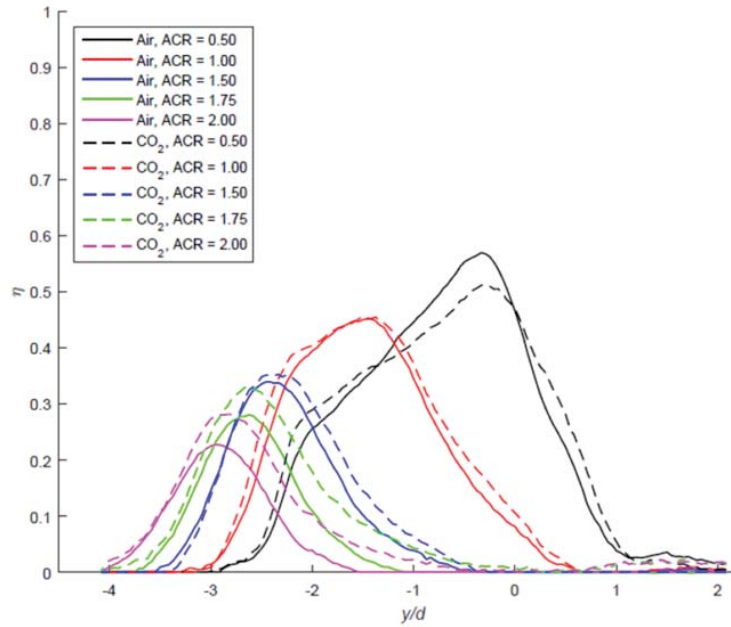


Figure 2.20. η distributions for multiple matched ACR values for both air and CO_2 [15].

III. Methodology

In order to accurately quantify the effects that each fluid property and transport ratio has on the adiabatic effectiveness on a flat plate geometry, it was important to have both accurate and precise data collection methods. This chapter will describe the facility (3.1), model design (3.2), experimental methods (3.3), and uncertainty analysis (3.4) used to ensure the highest quality data was collected.

3.1 Facility

The facility used for this research was the Air Force Research Laboratory (AFRL) Heat Transfer and Aerodynamics Lab, Test Cell 21. As seen in Figure 3.1, the tunnel is an open loop tunnel with a test section size of 36.8 cm by 40.6 cm. The flow is controlled via a flow control valve directly downstream of the single 50 hp blower. The air then passes through both a heater bank and a cold water chiller, although only the heater was used during testing. The heater was outfitted with a temperature control which was set to 327 K for all test runs, and validated with a J-type thermocouple inserted into the freestream in the test section. This ensured that freestream temperatures were held within 1 K during all testing days. The tunnel's

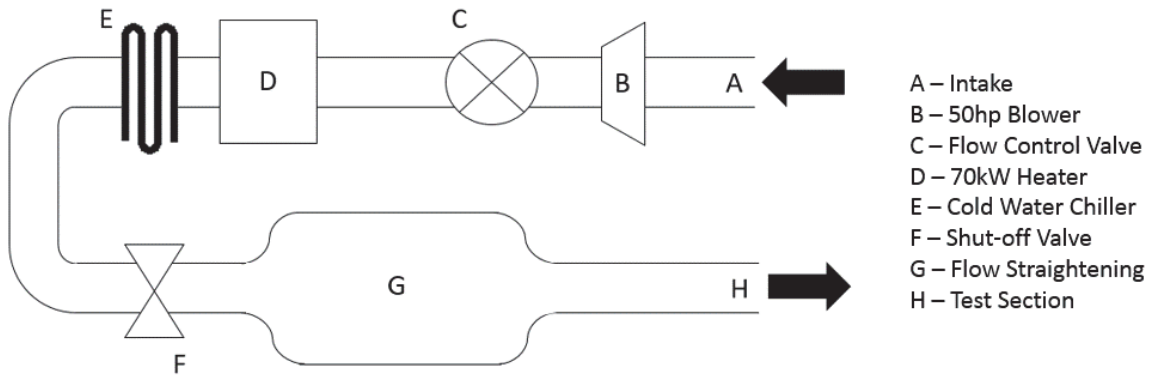


Figure 3.1. Wind tunnel schematic.

turbulence characteristics were quantified by Rutledge [11], and in the current test configuration, the tunnel exhibits very low turbulence levels at a turbulence intensity of 0.67%.

A schematic of the coolant flow path can be seen in Figure 3.2. The coolant required to run the experiments was routed from two different sources (A). When using air as a coolant, the supply was the shop compressed air line which was maintained at a pressure between 100 and 124 psi and a dewpoint lower than -40°C . When foreign gasses were required, the coolant was routed from various compressed gas bottles with a regulated pressure of 100 psi. The coolant flow was regulated using a manual pressure regulator (B) and then would pass through a counter flow, tube-in-tube heat exchanger (D) to control the coolant's temperature. The heat exchanger used fluid from a chiller which was set to maintain an ethylene glycol and water mixture at 255 K, and various valves on the heat exchanger allowed for coolant temperature control within 1 K.

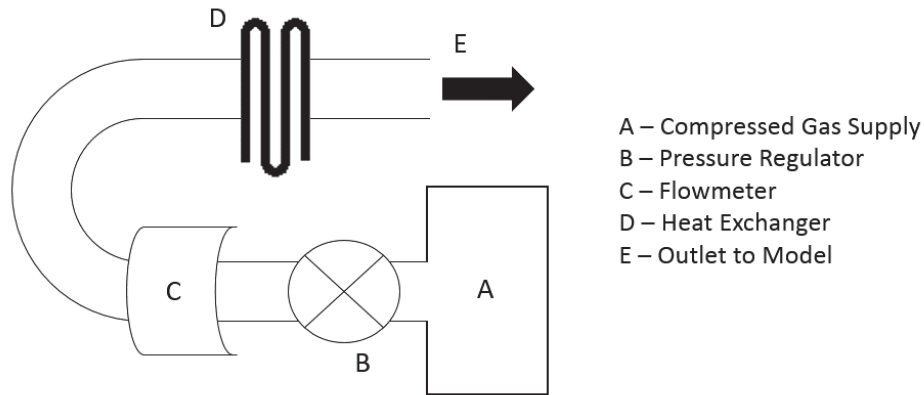


Figure 3.2. Coolant flow path schematic.

Much of the previous work conducted in this facility focused on studying cooling effectiveness on a scaled-up turbine blade leading edge. Both adiabatic and overall effectiveness distributions from various film cooling layouts on the leading edge of the airfoil have been collected in this tunnel. This meant that the camera position,

mounting holes, and data reduction code were all set up to view the leading edge models. In order to comply with facility capabilities as well as create the desired flat plate geometry from which to study, the choice was made to modify the test rig.

3.2 Model Design

The previous research was focused on leading edge film cooling which utilized a 0.089m diameter cylindrical leading edge model with a flat afterbody, seen installed in the test section as A in Figure 3.3. For ease, the new rig used this same design with an extended afterbody where the leading edge was extended forward in the tunnel, denoted as F in Figure 3.3. This allowed for a flat section to be in view of the IR camera without having to create new camera mounting points. Figure 3.4 shows a top-down view of the new rig installed in the tunnel using all previously installed mounting holes and the same camera position.

Figure 3.5 shows each component of the new test rig. The rig was constructed using a Plexiglas outer frame (green) and Last-a-Foam center sections of two different

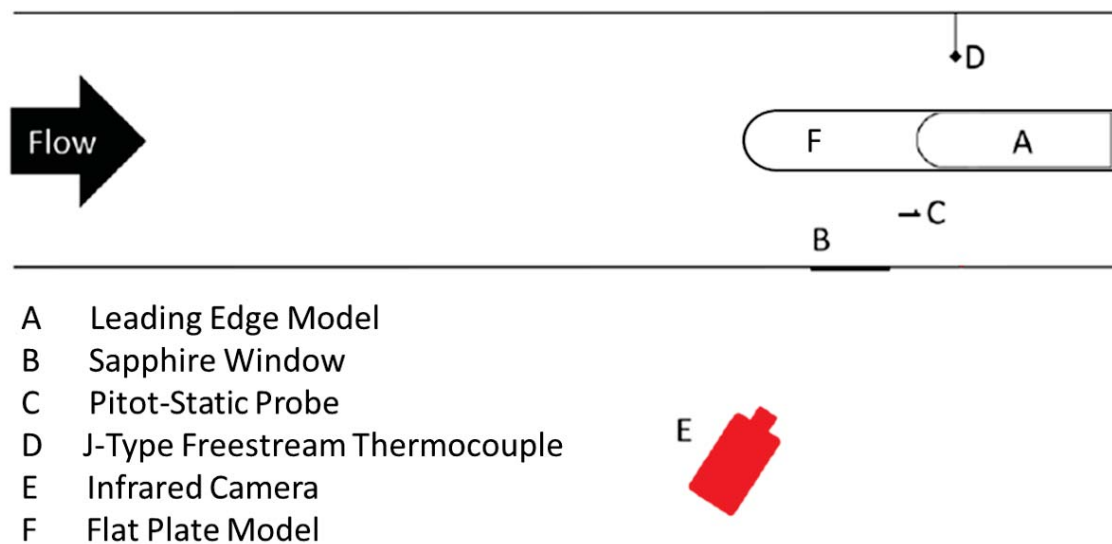


Figure 3.3. Test section schematic. Adapted from [15].

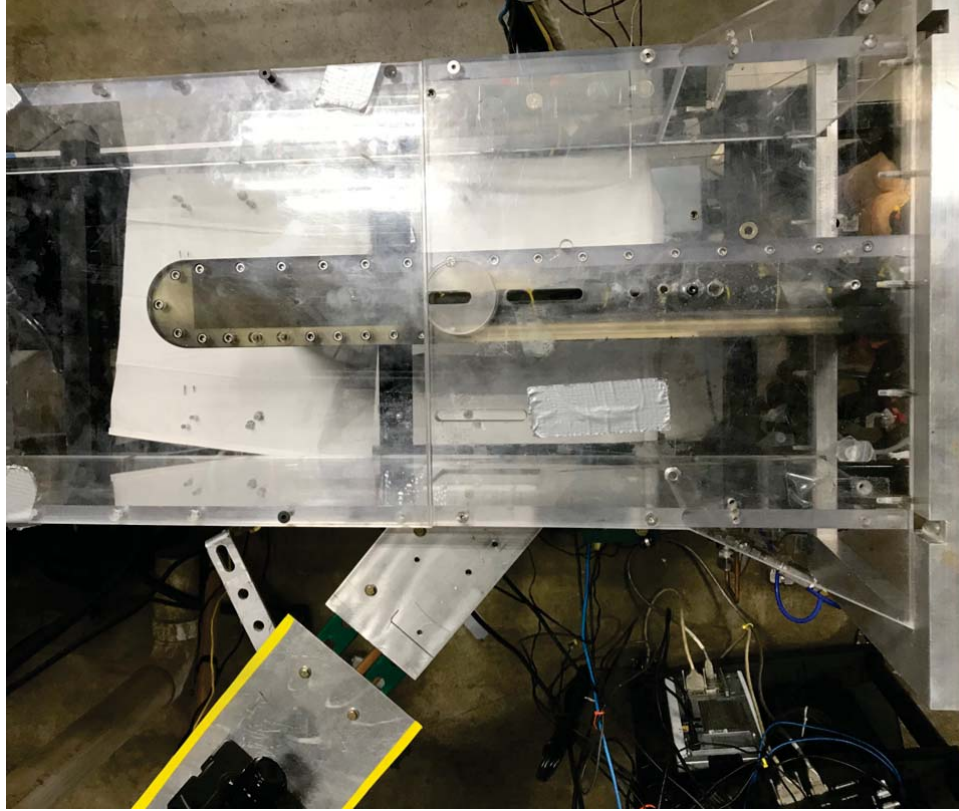


Figure 3.4. Top-down view of the new rig installed in the test section.

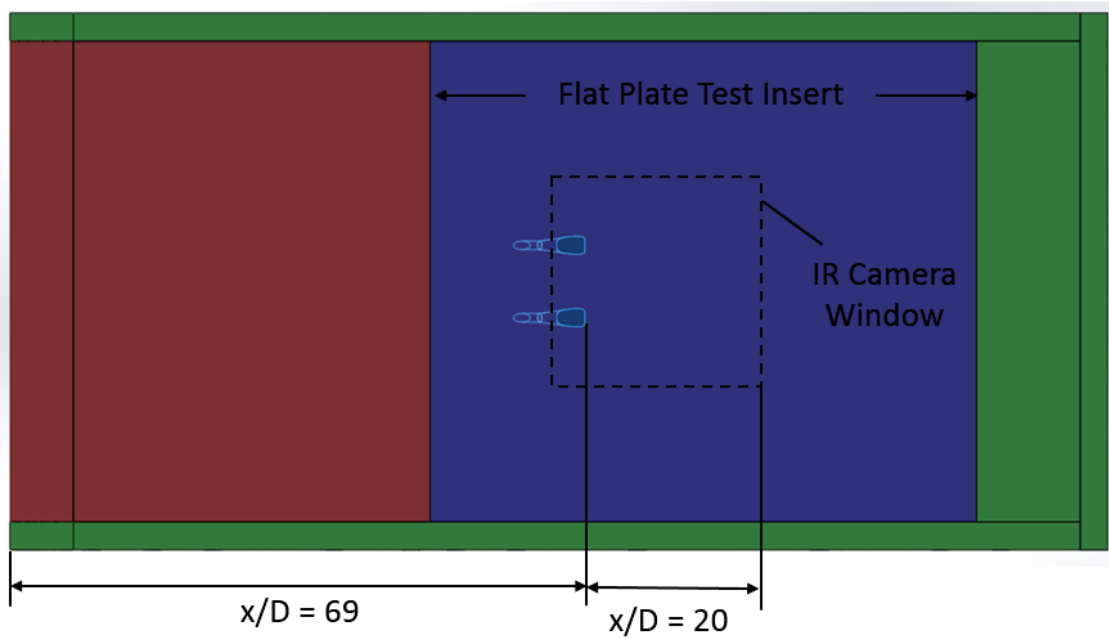


Figure 3.5. CAD model of the flat plate testing rig.

densities. All center section material upstream of the test section was constructed out of high-density Last-a-Foam, part number FR-7110, (red) to provide adequate structural integrity for the rig while maintaining a consistent transition to the test insert. The flat plate test insert itself was constructed out of low-density Last-a-Foam, part number FR-7106, (blue) in order to reduce the effect of conduction through the plate, thus requiring a smaller conduction correction during data reduction. The rig was designed to allow multiple flat plate test inserts to be exchanged without having to remove the entire rig from the tunnel.

The dimensions listed in Figure 3.5 were based on the non-dimensional distances used by Schroeder and Thole in the 7-7-7 hole analysis [13], and then multiplied by the metering diameter of the 7-7-7 hole, D , which for this study was $D = 5.81$ mm. The distance from the leading edge to the entrance of the cooling hole was $x/D = 69$, which is the same x/D as used in the 7-7-7 study. This was done to match the Re_x at the hole exit, which allows for an ‘apples to apples’ comparison to the data collected by Schroeder and Thole. The second dimension was the maximum visible length of the coolant plume constrained by the choice of not moving the IR camera placement. The area captured by the IR camera is outlined in Figure 3.5.

The first test insert constructed for the model featured a single 7-7-7 hole (described in Section 2.1.1), with a plenum spanning behind the entire section viewed by the IR camera, seen in Figure 3.6. The coolant was injected perpendicular to the test surface, impinging on the back side of the test plate. In this configuration, the coolant plume appeared to ‘fan-out’ at the far end of the camera window, as the impinging flow cooled the surface from inside the plenum through increased conduction, seen in Figure 3.7. This IR distortion would prove difficult to account for in the data reduction due to areas of increased conduction in the downstream region. Also, because the coolant had to travel across the entire length of the plenum before



Figure 3.6. First plenum constructed for the single hole test insert, mounted behind test surface.

exiting the cooling hole, the temperature of the coolant was difficult to control as the gas conducted heat from the relatively warmer test insert foam. It was also suspected that the design of the plenum caused an asymmetric coolant plume, which is seen in Figure 3.7. In order to reduce the difficulty of the conduction correction and increase the control of the coolant temperature, the decision was made to construct a new plenum.

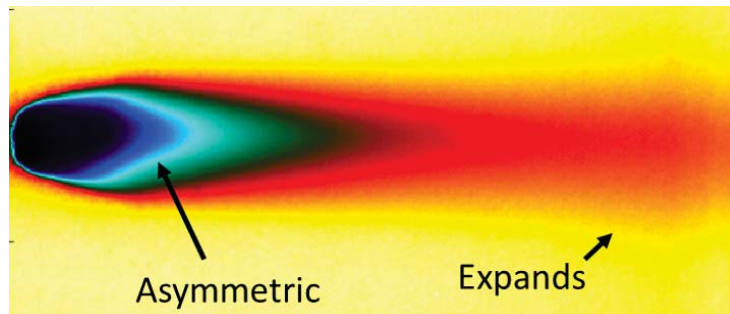


Figure 3.7. Raw IR image showing the coolant plume issues.

The new plenum design moved the coolant plenum outside the camera window, upstream of the coolant hole exit with the coolant being injected parallel to the freestream. The entrance to the coolant hole in Figure 3.8 lies at back of the plenum,

which flows coolant to the left onto the test surface. The plenum size was greatly reduced to shorten the purging time, but still large enough to ensure uniform flow out of the coolant hole. Now, the conduction through the test article would be reduced as the temperature inside of the test rig would become close to that of the freestream, and the uniform temperature behind the surface viewed by the IR camera would ensure uniform conduction through the test insert. Additionally, nine thermocouples were installed behind the IR view area in hopes to quantify the conduction in data reduction. These are also seen in Figure 3.8, installed at distances of $x/D = 3, 9, 15$, and separated laterally by $y/D = 3$. However, after this modification was completed, it was discovered that the leading edge rig used for the IR calibration did not produce accurate calibration curves for the flat plate rig as the new, extended geometry changed the radiation effects on the flat plate surface. In order to use the same rig for both data collection and IR calibration, it was decided to manufacture an entirely new test insert.

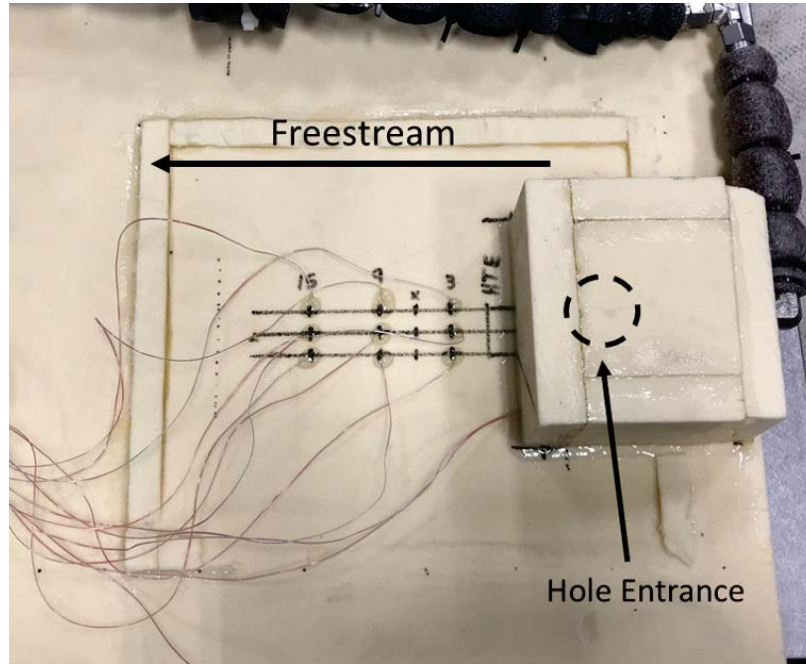


Figure 3.8. Second plenum constructed for the single hole test insert.

The second manufactured insert, seen in Figure 3.9, included two 7-7-7 holes, spaced at $y/D = 6$ to ensure that the presence of one hole did not affect the coolant plume of the other during testing. The bottom hole was used to conduct IR calibration runs, and thus was instrumented with thermocouples in its coolant plume, indicated by the red circles in Figure 3.9. The IR calibration process will be discussed in Section 3.4.1 The top hole was then used to gather data for the adiabatic effectiveness experiments. x/D values were defined as distances from the trailing edge of the top cooling hole, and y/D values were distances from the centerline of the top cooling hole, also indicated in Figure 3.9. To allow for an exact comparison between the IR data collection method and a future study using a Pressure Sensitive Paint (PSP)

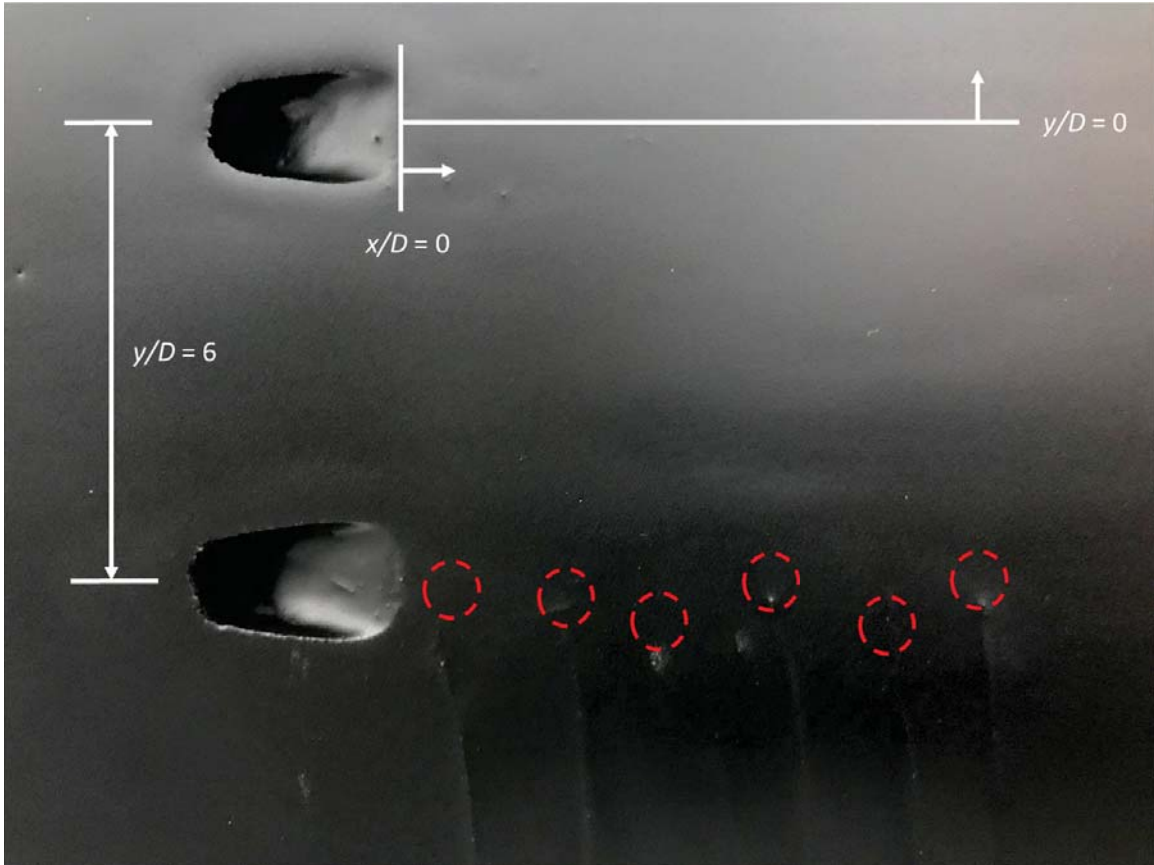


Figure 3.9. Test surface of two hole test insert with origin, spacing, and thermocouples indicated.

method, the foam surface was coated with a thin layer of Smooth-On epoxy. For the IR experiments, the epoxy was painted with a flat black spray paint, to equalize the emissivity of the surface. The plenums for the new two hole insert were constructed in a similar fashion as the second plenum for the single hole insert, which are pictured in Figure 3.10. The plenums were completely independent from one another and the coolant supplied to each was delivered via separate insulating tubing. Only one plenum was connected to the heat exchanger output at a time, so to switch between cooling holes, one would simply switch which tube was connected. The size of each plenum was kept as small as possible to allow for small purging times when switching the coolant gas, and their location upstream of the test surface allowed for a simplified conduction correction. This optimized two-hole insert was used to collect all the adiabatic effectiveness data presented in Chapter 4.



Figure 3.10. Plenums constructed for the two hole test insert.

3.3 Experimental Methods

In order to achieve the steady state test conditions listed in Table 3.1, the same set-up process was conducted each day before taking data. The first step before taking data was turning on the chiller to cool the ethylene glycol and water mixture for at least an hour prior to testing. This allowed the mixture to reach its steady state temperature of 255K. Next, the model was installed in the tunnel and the thermocouple extensions were plugged into their respective channels on the data acquisition system. After the model was secure, the tunnel and heater were turned on and run for at least 45 minutes to allow for all tunnel components to reach a steady state temperature. This was done not only for repeatability, but also for IR accuracy. It was found during testing that the temperature of the Plexiglas tunnel walls greatly impacted the IR calibration curve, so a consistent temperature during testing allowed for the same curve to be used across multiple days. The IR calibration process is described in Section 3.4.1. Data could then be collected once the tunnel and chiller temperatures reached their respective steady state values.

Table 3.1. Steady state test conditions during data collection.

Test Parameter	Value
$Re_{D,\infty}$	5000
$Mach$	0.03
T_∞	327 K
T_C	295 - 305 K
Tu	0.67%

The coolant flow rate was controlled manually by a pressure regulator, and monitored on the Omega FMA-1609A laminar flow element flowmeter, pictured in Figure 3.11. This flowmeter was rated to measure flows up to 50 standard liters per minute (SLPM) with a factory uncertainty of $\epsilon_{m_{dot}} = (0.8\% \text{ reading} + 0.1 \text{ SLPM})$. The

coolant then passed through the heat exchanger to lower its temperature, and then was delivered to the model via insulated tubing. The heat exchanger features many valves that can be adjusted to control the coolant temperature, and for all testing, the coolant temperature was held between 295 K and 310 K.



Figure 3.11. Omega flowmeter used to monitor flow rates.

Once the coolant flow rate was set, it was maintained for two minutes to allow the surface to reach a steady state condition. This equilibrium time was determined based on multiple equilibrium experiments, where the coolant flow rate was suddenly changed from a low value to a high value and data was taken every 10 seconds. Figure 3.12 shows data from one of these experiments. The η_{app} distribution along the $y/D = 0$ line is plotted at every time step, over the course of eight minutes. The lines collapsed very quickly into a tight group, so the difference between each consecutive timestep ($\eta_i - \eta_{i-1}$) was plotted to show how much the values differ over time. The resulting plots are seen in Figure 3.13. The left plot shows all the difference

lines at an elapsed time of two minutes, and the right shows all difference lines after the full eight minute test. One can see that after two minutes, the difference lines become ‘bounded’ between $\Delta\eta = \pm 0.03$. This is less than the expected uncertainty of ± 0.04 , so two minutes was chosen as an appropriate equilibrium time.

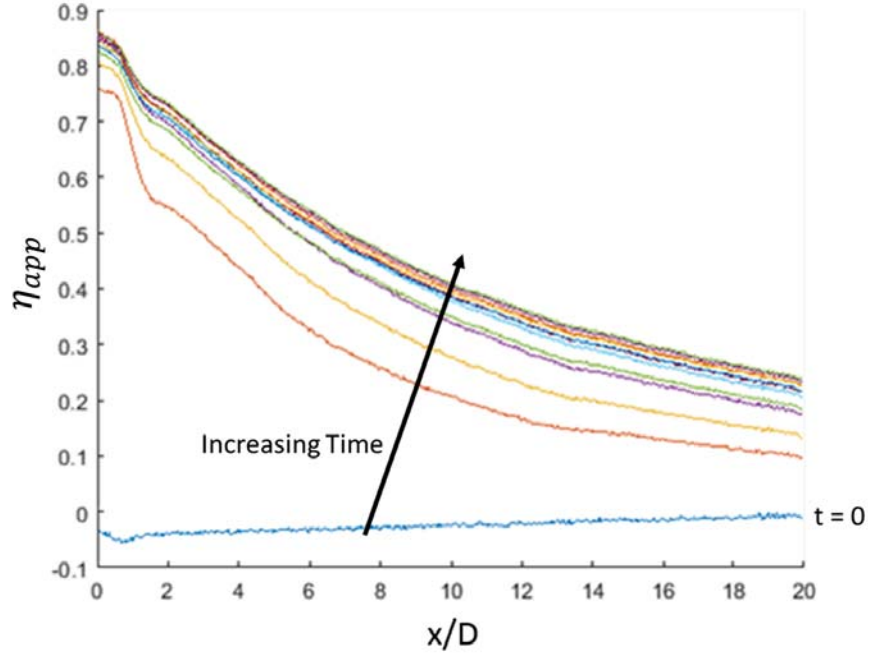


Figure 3.12. η_{app} distribution along $y/D = 0$ over the course of eight minutes.

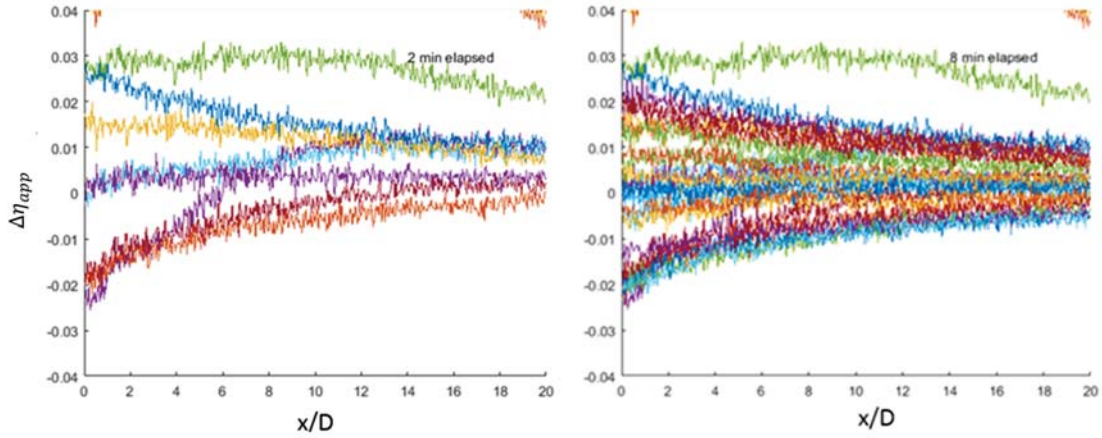


Figure 3.13. Difference in η_{app} distributions after two minutes (left) and eight minutes (right).

3.3.1 Data Acquisition and Reduction

Most of the data acquisition process and code was the same as described by Wiese [15] in his Section 3.2 and 3.3. Essentially, the freestream conditions were determined by a pitot-static system and a freestream thermocouple. For this research, the freestream Reynolds number based on the cooling hole diameter, or Re_D , was set to a value of 5000. The LabView code then calculated the coolant flow rates required to achieve the desired ratio values. These flow rates were then achieved by adjusting the manual pressure regulator and watching the Omega flowmeter display. The coolant temperature is collected by a J-type thermocouple which is secured at the entrance of the cooling hole in the plenum. Only the thin wires and weld bead were exposed in the hole entrance to minimize disturbances to the cooling flow. When the ‘Take Data’ button in the LabView GUI was pressed, the LabView program wrote the current temperatures and flow rates to a text data file and also triggered the IR camera to capture the current image. The data points in the text file were later matched to the corresponding IR images by the Matlab data reduction code.

To account for the camera being mounted at an angle from the test surface, a spatial calibration was done to resize the captured IR images to the correct physical size. This process used the same code developed by Wiese [15], and is described in detail in his Section 3.5. The code uses an IR image of the test surface where a grid of pins was secured beneath the top cooling hole, seen in Figure 3.14. Each pin location is marked by the user and the code outputs coefficients for a curvefit that is used to flatten the image for analysis.

Once the images are resized from the spatial calibration, the raw counts data is converted to actual surface temperatures through use of the IR camera calibration, described in Section 3.4.1. This calibration yields an equation for the surface temperatures as a function of counts output by the IR camera. Once the surface

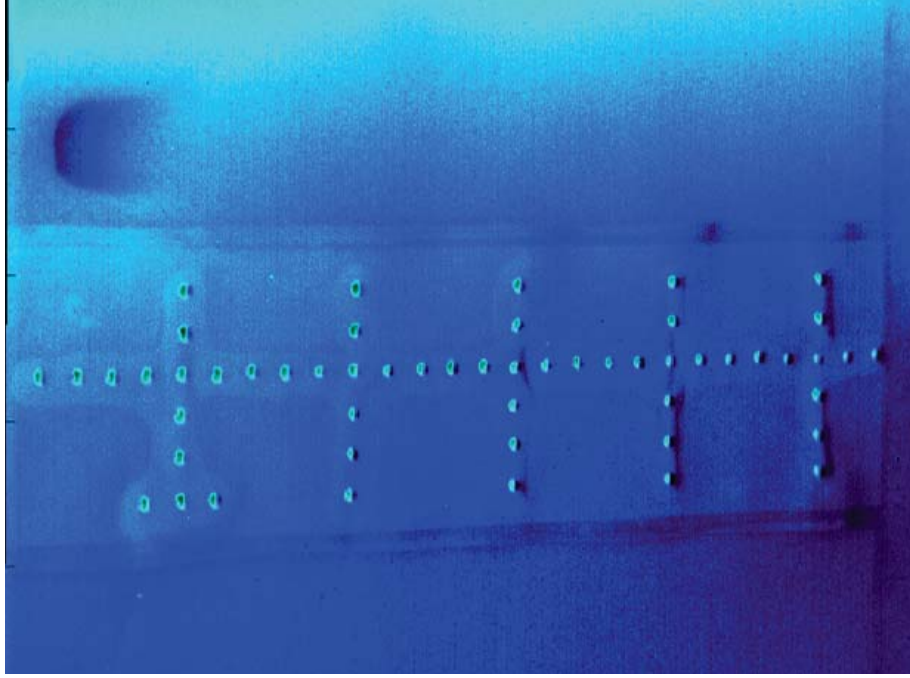


Figure 3.14. Raw IR image of the spatial calibration set-up.

temperatures are calculated, η can then be calculated for the entire surface using the freestream and coolant temperatures for non-dimensionalization. However, this value does not account for conduction through the test article.

To correct for this, the conduction correction used by Williams et al. [18] (also presented in Section 2.2.2) was implemented. The previously calculated η values were used as the apparent adiabatic effectiveness, or η_{app} , values.

$$\eta = \frac{\eta_{app} - \eta_0}{1 - \eta_0} \quad (3.1)$$

The η_0 values were calculated by averaging three η_{app} data points above the coolant plume and three below. Because these points are outside the coolant plume, the only cause for a lower temperature is conduction into the test rig. η_{app} and η_0 were then used in Equation 3.1, resulting in the corrected η values presented in Chapter 4.

3.4 Uncertainty Analysis and Calibration

The uncertainty analysis utilized the methods presented by Kline and McClintock [10]. This method defines the overall uncertainty in an arbitrary quantity, ϵ_ζ , as a function of the uncertainty of each measured quantity, ϵ_{ξ_i} . These uncertainties are then multiplied by the partial derivative of ζ with respect to each measured quantity, ξ_i and then combined in a root-sum-square method, as seen in Equation 3.2.

$$\epsilon_\zeta = \sqrt{\left(\frac{\partial\zeta}{\partial\xi_1}\epsilon_{\xi_1}\right)^2 + \left(\frac{\partial\zeta}{\partial\xi_2}\epsilon_{\xi_2}\right)^2 + \dots + \left(\frac{\partial\zeta}{\partial\xi_n}\epsilon_{\xi_n}\right)^2} \quad (3.2)$$

3.4.1 Freestream, Coolant, and Gas Property Uncertainties

Because this research took place in the same facility and used the same data reduction code as the research done by Wiese [15], details for the freestream, coolant, and gas property uncertainties can be found in his Section 3.9. Any temperature collected using a thermocouple carried a standard uncertainty of $\pm 0.3K$. Uncertainty in the freestream Reynolds number was calculated to be $\pm 3\%$ of the targeted value of $Re_D = 5000$. The uncertainties of the coolant gas properties were determined using both the ideal gas law as well as linear interpolation of tabulated values and can be seen in Table 3.2.

Table 3.2. Uncertainties for the coolant gas properties [15].

$\epsilon_{\rho_c/\rho_\infty}$	$\epsilon_{c_{P,c}/c_{P,\infty}}$	$\epsilon_{\mu_c/\mu_\infty}$
0.2%	0.7%	2.8%

As listed previously in Section 3.3, the factory uncertainty of the flowmeter depends on the value of the flow rate. As the flow rate increases, the uncertainty decreases as seen in Figure 3.15. All data points in this study required flow rates greater than 5 SLPM, thus the maximum uncertainty in the flow rate was $\pm 3\%$.

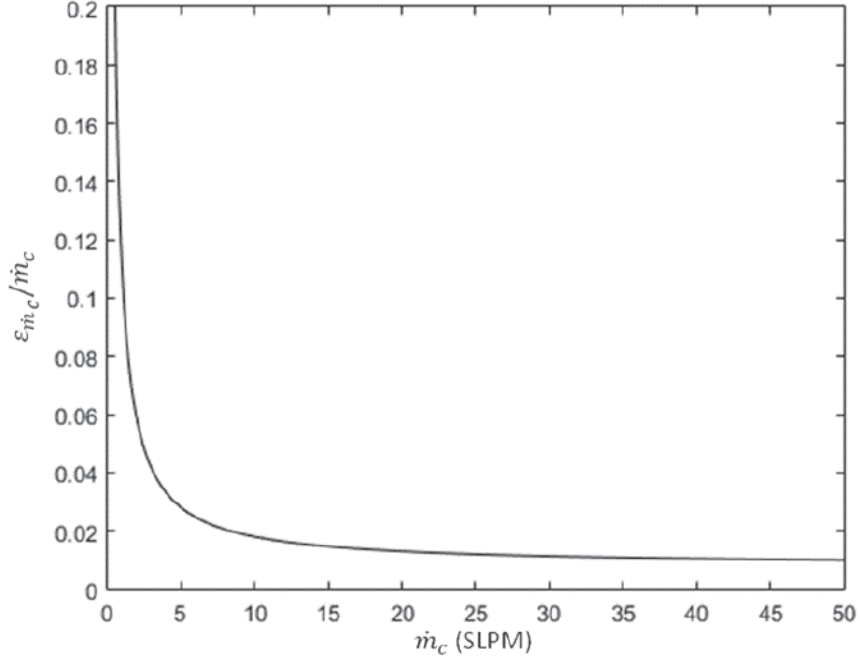


Figure 3.15. Uncertainty in the coolant flow rate as a function of the reading [15].

3.4.2 IR Camera Calibration

The IR camera calibration utilized the lower hole on the constructed test insert, seen in Figure 3.9. In order to adjust the surface temperature, coolant was flowed over the installed thermocouples at increments of 5 SLPM up to 20 SLPM, and then by increments of 10 up to 50 SLPM, which is the maximum flow rate that the Omega flowmeter can handle. At each increment, data was taken after a constant flow rate for two minutes as required for equilibrium (described in Section 3.3.1). After the maximum flow rate was reached, the increments were completed in reverse order down to 0 SLPM to detect any hysteresis, as this would indicate a bad calibration. The importance of the tunnel wall temperature, described later in this section, was shown by the presence of hysteresis in previous calibrations. Once all the data points were collected, the dataset was run through a Matlab reduction code, written by Wiese [15] and described there in Section 3.4. The code used a marking pin on the surface of the test insert as a reference, and then backed out each thermocouple's location using

a provided text file. The text file contained each thermocouple's location from the marking pin in terms of how many pixels in the x and y directions. At each location, the code averaged a 2x2 pixel area to use as the measured counts value, which was then matched to the temperature recorded by the respective thermocouple to create a data point. Because a small area was averaged, it was important to place the thermocouples in areas of relatively small temperature gradients. If a high temperature gradient was present, even a one pixel error in the thermocouple's location could yield a 1K error in the temperature reading. All data points were then plotted and a Matlab polyfit was applied to the data points to generate the IR calibration curve. An example of a calibration dataset and curvefit can be seen in Figure 3.16. Because six thermocouples were used, each increment of the coolant flow produced six data points, resulting in a complete coverage of the observed temperatures. The uncertainty, ϵ , listed in Figure 3.16 was calculated using the standard deviation of the differences in temperature (K) between each data point and the generated calibration curve. For the final ϵ value, the standard deviation was multiplied by a Student T factor to account for a smaller number of data points.

Multiple IR calibrations were done during the course of testing to ensure that the curve was stationary. As seen in Figure 3.17, the four calibrations done with the flat plate model essentially resulted in the same calibration curve, with $\pm 0.5\text{K}$ variations. The fifth line, the Leading Edge line, was generated from the calibration rig previously used in the facility when testing on the leading edge. Figure 3.17 proves that the flat plate rig requires its own calibration method as the radiation exchange between the tunnel walls and the new rig is different from the leading edge experiments. Differing geometries have different radiation view factors which ultimately change the net heat flux due to radiation and thus alter the IR calibration curve.

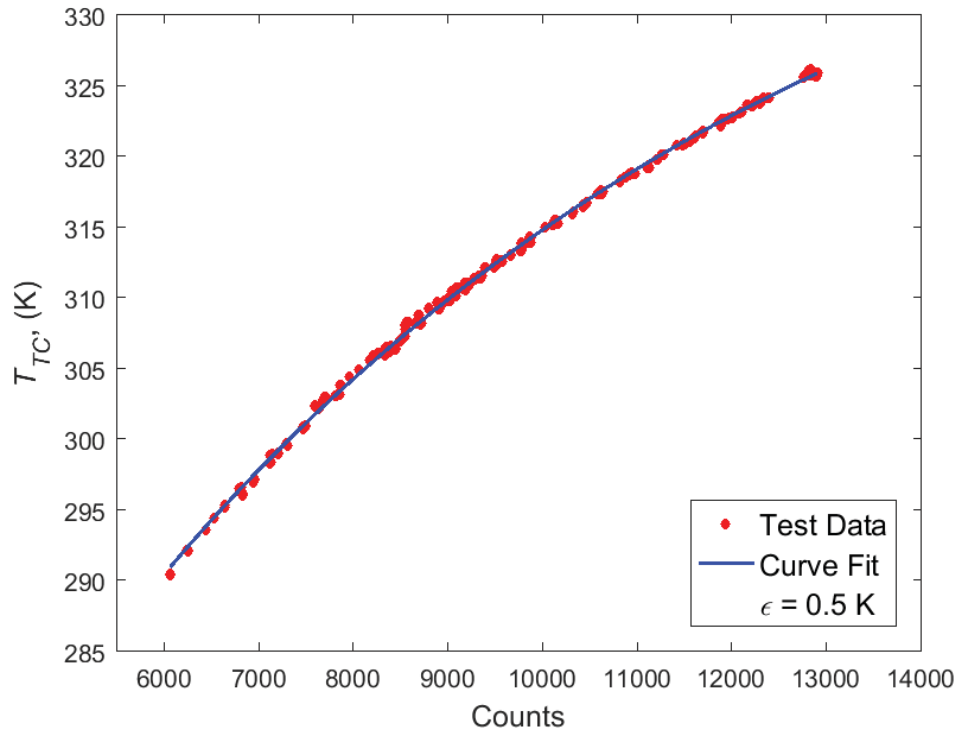


Figure 3.16. Generated IR calibration curve.

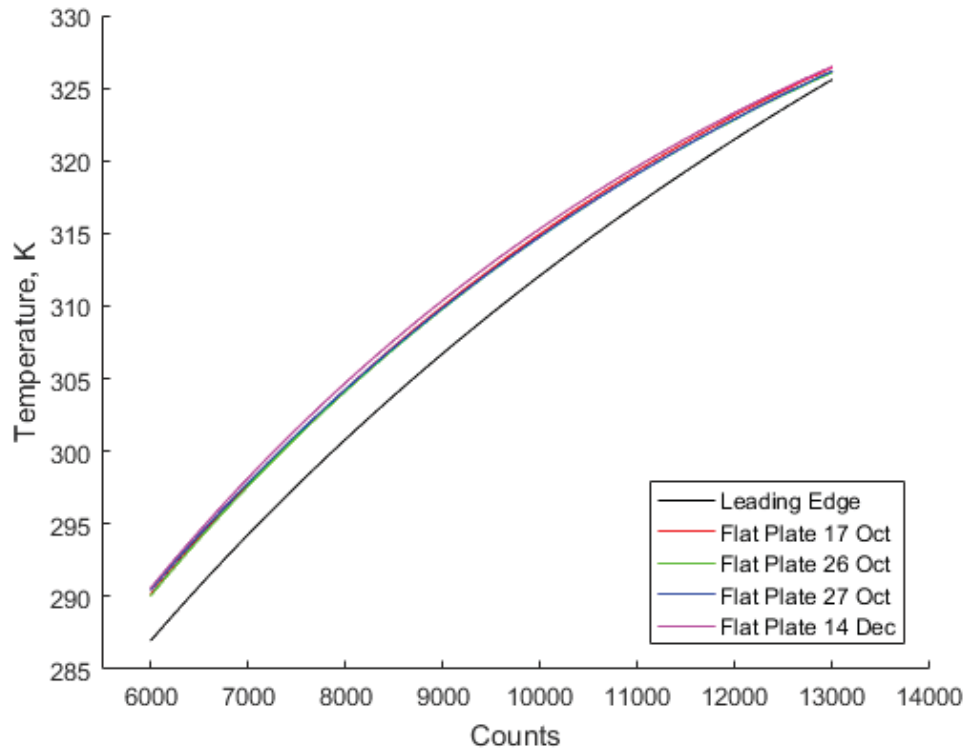


Figure 3.17. IR calibration curves from the flat plate and leading edge calibration rigs.

During the IR calibration process, it was also found that the temperature of the Plexiglas tunnel walls influence the calibration curve. As seen in Figure 3.18, when the tunnel walls were cool, the calibration curve drops significantly at higher count values. The close grouping of calibration curves as seen in Figure 3.17 was achieved by running the tunnel with the heater bank set to a constant temperature for all days of testing. This reduces the effect of ambient temperature changes, and essentially allows for the same calibration to be used over multiple days of testing.

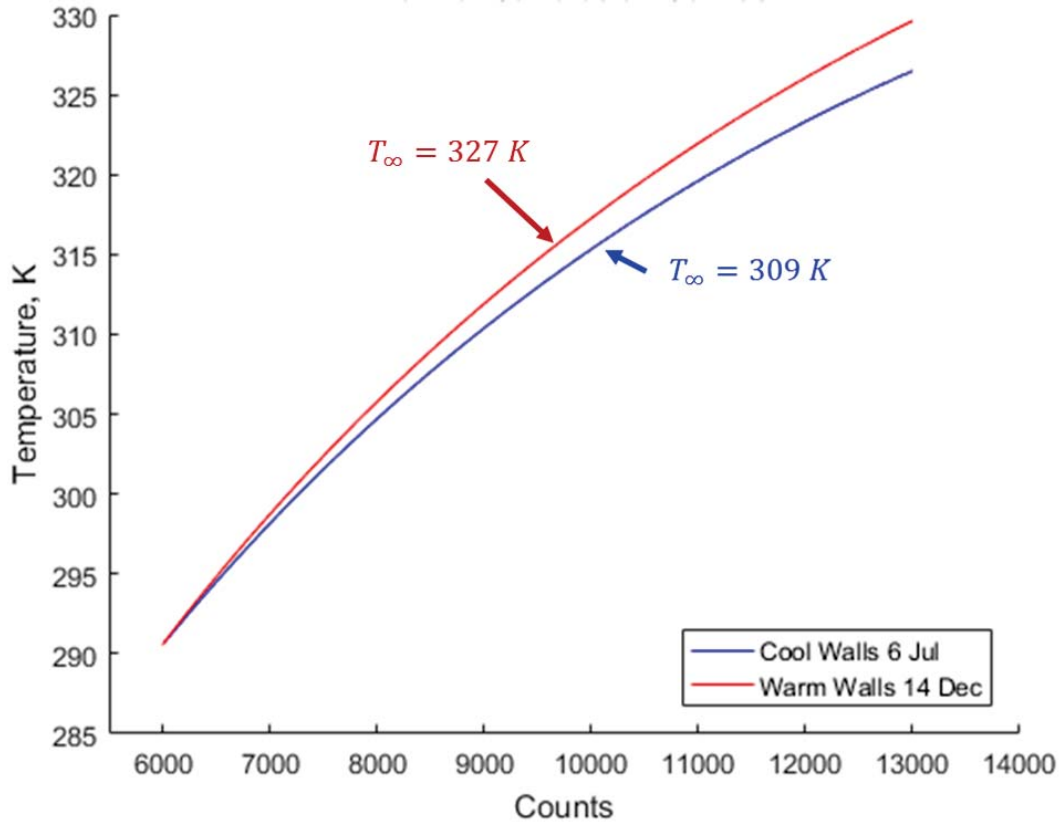


Figure 3.18. IR calibration curves from two different Plexiglas wall temperatures.

3.4.3 Adiabatic Effectiveness Uncertainty

The total uncertainty in η was calculated using Equation 3.2, and all uncertainties previously listed in this section. First, the uncertainties of η_{app} and η_0 were calculated using the standard thermocouple uncertainty for the directly measured temperature

readings, and the calculated IR uncertainty for the surface temperature uncertainty. Then, the uncertainty in η was calculated using η_{app} and η_0 and their respective uncertainties as the measured quantities. The total η uncertainty was calculated at two surface locations of two tested data points. The two test points were selected to represent the lowest and highest ΔT values, where $\Delta T = T_\infty - T_c$. The lowest ΔT during testing was 22, and the highest 32. To find the maximum and minimum η uncertainty values for the study, the surface locations were selected at the points of highest and lowest surface temperatures, resulting in the values seen in Table 3.3. As expected, the uncertainty was largest in areas of low η and low ΔT , reaching a maximum value of 0.04. In areas of high η and high ΔT , the uncertainties fell to as low as 0.02.

Table 3.3. Maximum and minimum total uncertainties for the collected η dataset.

$\Delta T = 32 \text{ K}$		$\Delta T = 22 \text{ K}$	
<u>Lowest T_s</u>	<u>Highest T_s</u>	<u>Lowest T_s</u>	<u>Highest T_s</u>
$\varepsilon_\eta :$ 0.019	0.026	0.030	0.039

IV. Results and Discussion

The results of the completed experiments will be discussed in this chapter in order to accomplish the objectives of this research. The initial objective evaluated was to determine if the use of helium resulted in any temperature separation effects with the new geometry configuration. This will be discussed in Section 4.1. The main objective was to analyze the performance of each scaling parameter used for low temperature gas turbine film cooling testing, and determine the importance of property ratios other than the density ratio. This will be discussed in Section 4.2.

4.1 Temperature Separation Results

The initial objective of this study was to determine if helium (He) was a suitable option to use during low temperature testing. As seen in Section 2.3.2, when helium was previously used on a leading edge model, surface temperatures deviated as much as 10K for certain flow conditions, even with T_c equal to T_∞ . To investigate this phenomena on the new flat plate rig, a VR sweep was completed to examine a range of flow velocities using helium as the coolant gas. The goal was to match the coolant temperature with the freestream temperature, so any surface temperature differences would only be a result of the energy separation phenomena. Since the heat exchanger controls must be adjusted manually, coolant temperatures were not exactly that of the freestream, but were held within 1K. Because of conduction through the test insert, the surface temperatures outside the coolant plume were consistently 1-2K cooler than the freestream, so Figure 4.1 shows the difference in measured surface temperature (T_s) from an average surface temperature outside the coolant plume ($T_{outsideplume}$). As seen in the figure, there were deviations indicated with maximum values of $\pm 3K$. The points of highest deviation exist in small areas in line with the

upper and lower edges of the cooling hole, while deviations directly downstream on the centerline only reach values as high as 2K at the $VR = 0.5$ case. It is important to note however, that the coolant temperature in the $VR = 0.5$ case was already almost 0.5K warmer than the freestream, which resulted in the coolant falsely increasing the surface temperature. Subtracting 0.5K from any deviation seen in the $VR = 0.5$ case yields a maximum temperature deviation of 1.5K along the centerline.

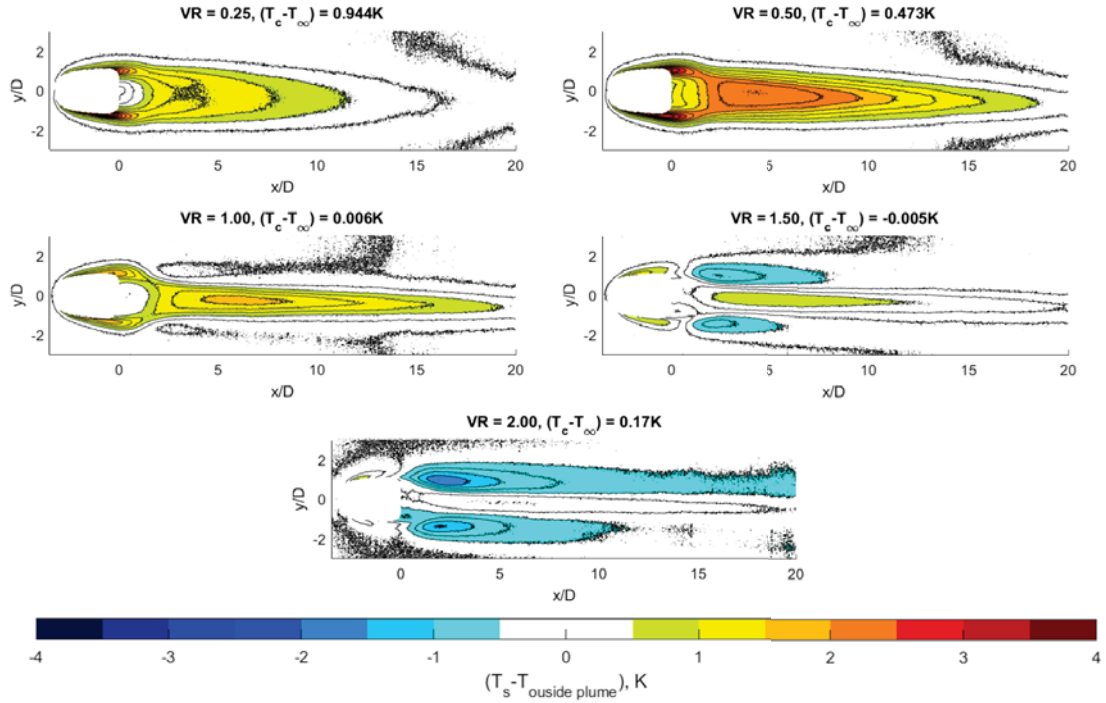


Figure 4.1. Difference between surface and freestream temperatures at various VR values using helium as coolant.

The impact of any temperature separation can be quantified as a fraction of the expected ΔT between the coolant and freestream gas because the surface temperatures are ultimately nondimensionalized by the ΔT . During the majority of testing, this ΔT value was approximately 30K. This results in a worst case scenario of a 5% deviation of the ΔT along the centerline, or 0.05 drop in η due to temperature separation effects. Because these effects were only seen at a small range of flow rates and if present will only impact the η results in the area of interest by a maximum drop

of 0.05, it was decided that helium was a suitable gas to use for this geometry.

4.2 Adiabatic Effectiveness Results

In order to fully evaluate the ability of each of the five coolant flow rate parameters (M , I , ACR , VR , and ReR) to scale adiabatic effectiveness, each parameter was matched at the values listed in Table 4.1 using air, argon (Ar), carbon dioxide (CO₂), helium (He), and nitrogen (N₂) as the coolant gas. Every parameter was incrementally matched at the listed values. Test runs of M , I , and ReR were halted prematurely when using helium because the matched values required a flow rate higher than the 50 SLPM limit of the flowmeter. For this reason, helium data is absent from some of the presented results.

To minimize the impact of any potential temperature separation effects, data was compared using methods that either avoided the problem regions or minimized their effect. These methods included η contour plots, centerline η values, and area averaged η values ($\bar{\eta}$). The centerline η values avoid the most common problem regions above and below the plume's centerline. The $\bar{\eta}$ was calculated using the area bounded by $y/D \pm 3$ and x/D from 3 to 20. This area is much larger than any of the problem regions seen in Figure 4.1, thus dampening any change to the overall $\bar{\eta}$. Also, the effect on η was minimized by using a ΔT between the coolant and freestream temperatures that was much larger than the magnitude of any observed temperature separation effect.

Table 4.1. Matched values collected for each flow rate parameter.

Coolant Flow Rate Parameter	Matched Values
M , VR , ReR , and ACR	0.17, 0.25, 0.34, 0.50, 0.67, 1.00, 1.34, 1.50, 2.00
I	0.11, 0.23, 0.25, 0.45, 0.50, 0.67, 0.90, 1.00, 1.50, 2.00

4.2.1 Blowing Ratio, M

The first parameter analyzed was the blowing ratio due to its widespread use in the literature. Figure 4.2 shows that at a low value of 0.25, M did not scale well between all five gases. Helium outperformed the other four gases, but the differences between the rest are better seen along the centerline, as plotted in Figure 4.3. The large spread of values indicates the inability of M to scale η . However, Figure 4.3 was the first indication of the importance of ACR , as the η profiles on the line plot were ordered with increasing ACR values; helium being the highest at 1.29, and argon being the lowest at 0.13. Because ACR is simply M multiplied by the specific heat ratio between the coolant and freestream, the increase in η between argon and helium can be attributed to an increase in $C_p R$.

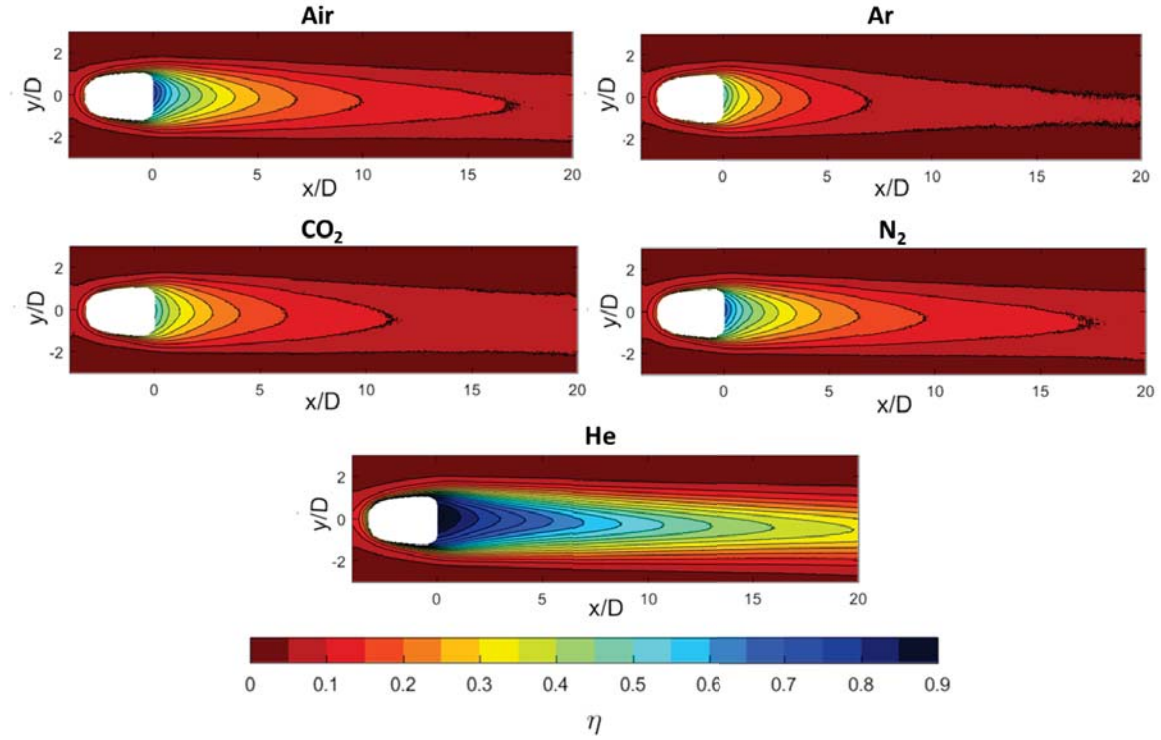


Figure 4.2. Contours of η at the matched $M = 0.25$ case.

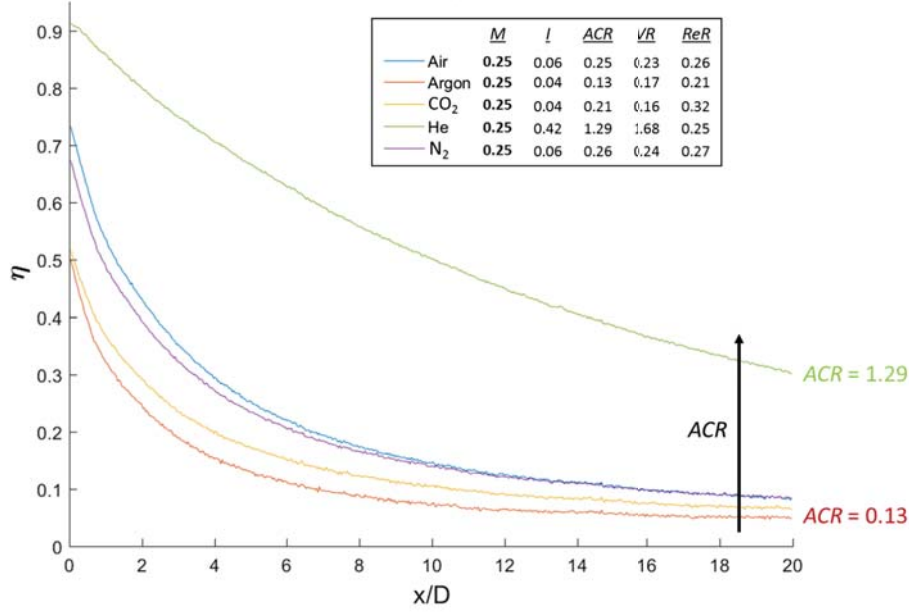


Figure 4.3. Distribution of η along $y/D=0$ at the matched $M = 0.25$ case.

This dependence on $C_p R$ is less evident at higher flow rates, as seen in Figure 4.4 where M was matched at a value of 1.5. This value of M required a helium flow rate above 50 SLPM, thus the absence of the helium line. As before with $M = 0.25$, argon yielded the lowest η values, but the CO₂ values were higher than air or N₂ despite having a lower ACR . This break in dependence on $C_p R$, could be a result of jet separation effects. As shown previously in the literature, I has been proven to be a good indicator of jet separation and all four values of I in this case were above 1.0, with air and nitrogen being above 2.0. The approximate separation value will be discussed later with the I results.

It should be noted that if the argon data was removed from Figure 4.4 and the helium and argon data from Figure 4.3, one may be inclined to conclude that M is an appropriate scaling parameter to use, especially at high flow rates for this geometry. This conclusion is negated by the use of argon and helium which show the inability of M to scale gases with drastically different properties than the freestream.

The effects of jet separation on M can be clearly seen in Figure 4.5 as the $\bar{\eta}$ distributions peak and then decline for the four heavier gases. In the region of $M < 1.0$, the gases are again ordered with increasing $C_p R$. At values of $M > 1.0$, the air/N₂ and CO₂ trends switched places, as seen previously in Figure 4.4. Also listed in Figure 4.5 are the average values of DR and $C_p R$ for each gas, calculated using the individual DR and $C_p R$ at every data point. The actual values of DR varied by ± 0.1 and the $C_p R$ values varied by ± 0.2 . The inconsistencies of DR testing are prevalent again as CO₂ and argon have roughly the same DR , but very different $\bar{\eta}$ values. It is also important to note that the shape of the $\bar{\eta}$ distributions is very similar to the data collected by Schroeder and Thole [13] seen in Figure 2.14. This similarity between data sets confirmed that the hole was manufactured correctly and was producing similar data to other 7-7-7 holes.

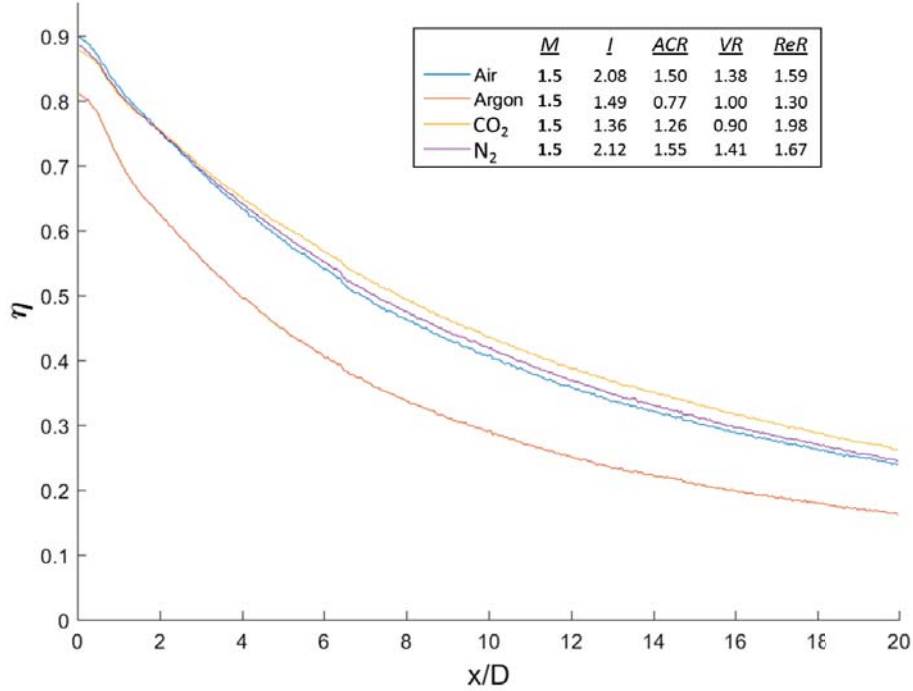


Figure 4.4. Distribution of η along $y/D=0$ at the matched $M = 1.50$ case.

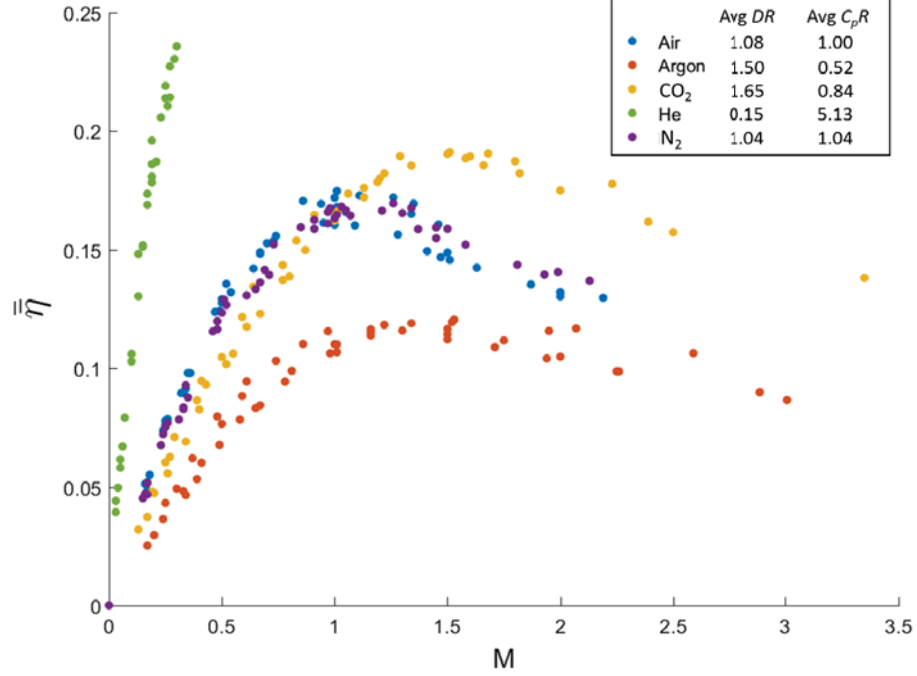


Figure 4.5. Area-averaged η values ($\bar{\eta}$) at all resulting M values. DR and $C_p R$ values listed are averages among all test points.

4.2.2 Momentum Flux Ratio, I

Due to the potential for jet separation having an effect on the data, the next parameter analyzed was the momentum flux ratio. At low values, such as the $I = 0.25$ case seen in Figure 4.6, I appears to perform better than M in Figure 4.3, but still does not scale all five gases to the same η values. It is noted again that the highest and lowest ACR values correspond respectively with the highest and lowest η values, however, the CO_2 line breaks this trend and falls slightly below air and N_2 . The difference between the ACR values for air, CO_2 , and N_2 was 0.02, which could explain the close grouping of the three η profiles, again showing the importance of the $C_p R$. As with the M results, if the argon and helium data were removed from Figure 4.6, one may be inclined to conclude that I is the appropriate scaling parameter to use for this geometry.

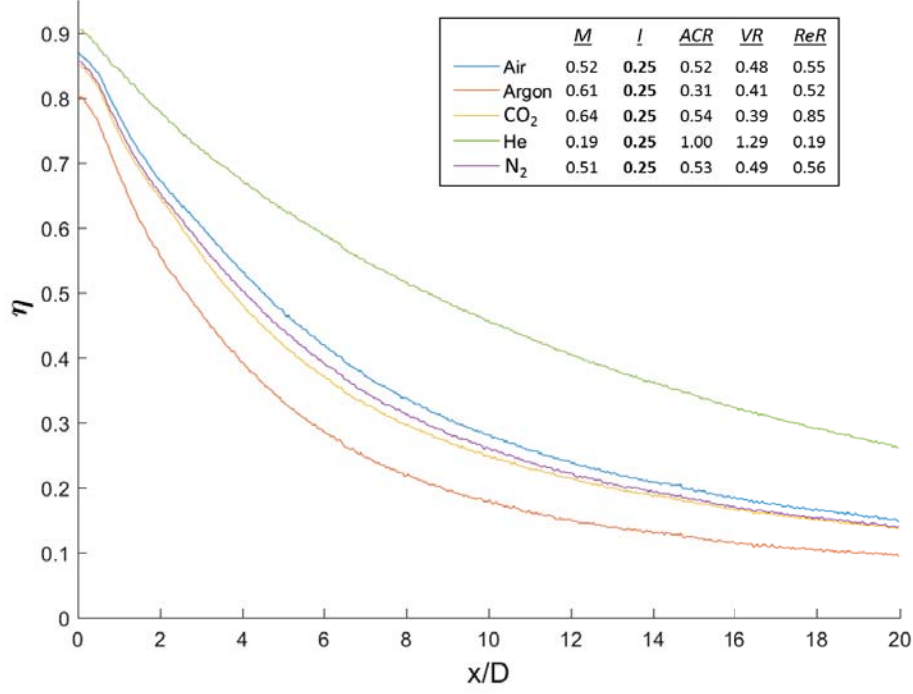


Figure 4.6. Distribution of η along $y/D=0$ at the matched $I = 0.25$ case.

The close grouping of air, CO₂, and N₂ was also seen at higher I values. Figure 4.7 shows the results of the $I = 1.50$ case, where the three gases yielded η values that were closely grouped across the entire x/D span. This grouping can also be attributed to a small difference in the ACR values, which in this case was 0.04 between air and CO₂. If argon was removed from Figure 4.7, I would appear to be a better choice than M as the η profiles between air, CO₂, and N₂ stay consistently close to one another over all flow rates. Coupled with the previous conclusions of other studies that I is a good indication of jet separation, I would be an easy choice for the best scaling parameter in the absence of helium and argon data.

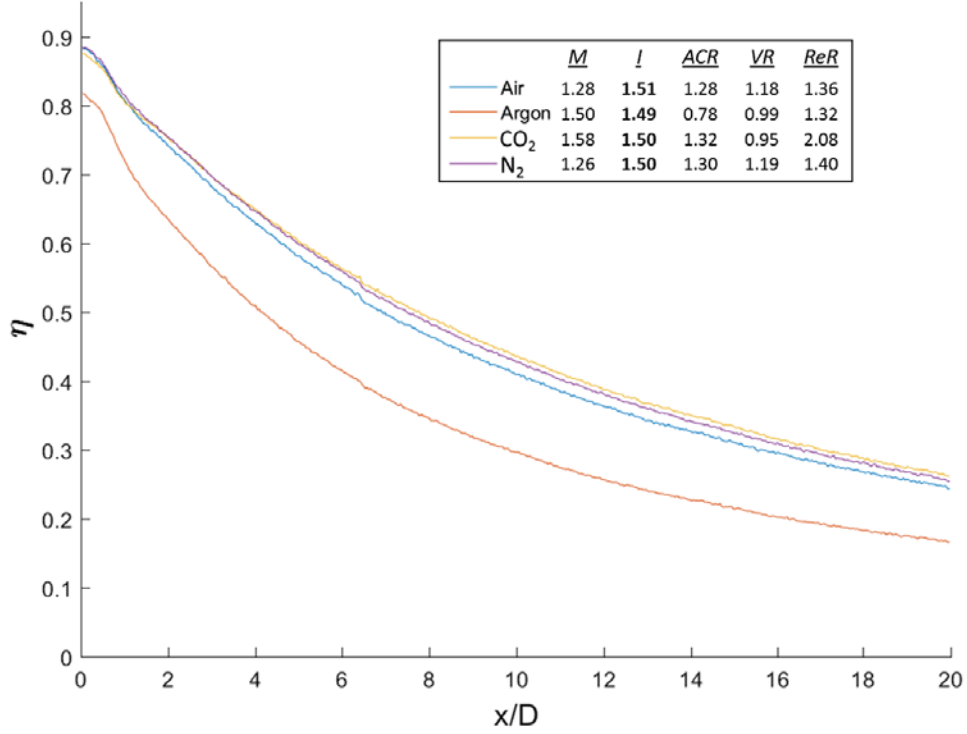


Figure 4.7. Distribution of η along $y/D=0$ at the matched $I = 1.50$ case.

Although I did not scale the η values between all the gases, it was found to be an important parameter in terms of predicting the degree of jet separation. This relationship is consistent with previous findings in the literature, and is most evident when analyzing the $\bar{\eta}$ distributions in Figure 4.8. When I is matched, the peak value of $\bar{\eta}$ for each gas occurs at the same value of $I \approx 1.2$, and the shape of each distribution is similar. This similarity was expected as the same hole geometry was utilized for all testing. Because jet separation is a momentum driven fluid effect, the jet leaving a given hole geometry would be expected to begin to separate when the momentum of the fluid reaches a critical value unique to every geometry. Jet separation is independent of the specific heat of the fluid, resulting in different magnitudes of $\bar{\eta}$ for each gas. It is important to note that after each distribution reaches its max at $I \approx 1.2$, $\bar{\eta}$ does not drop to zero. This result indicates that part of the coolant

plume remains attached to the surface at all flow rates, and can be attributed to the ability of fan-shaped, laid-back hole geometries to slow part of the jet down enough to remain attached to surface.

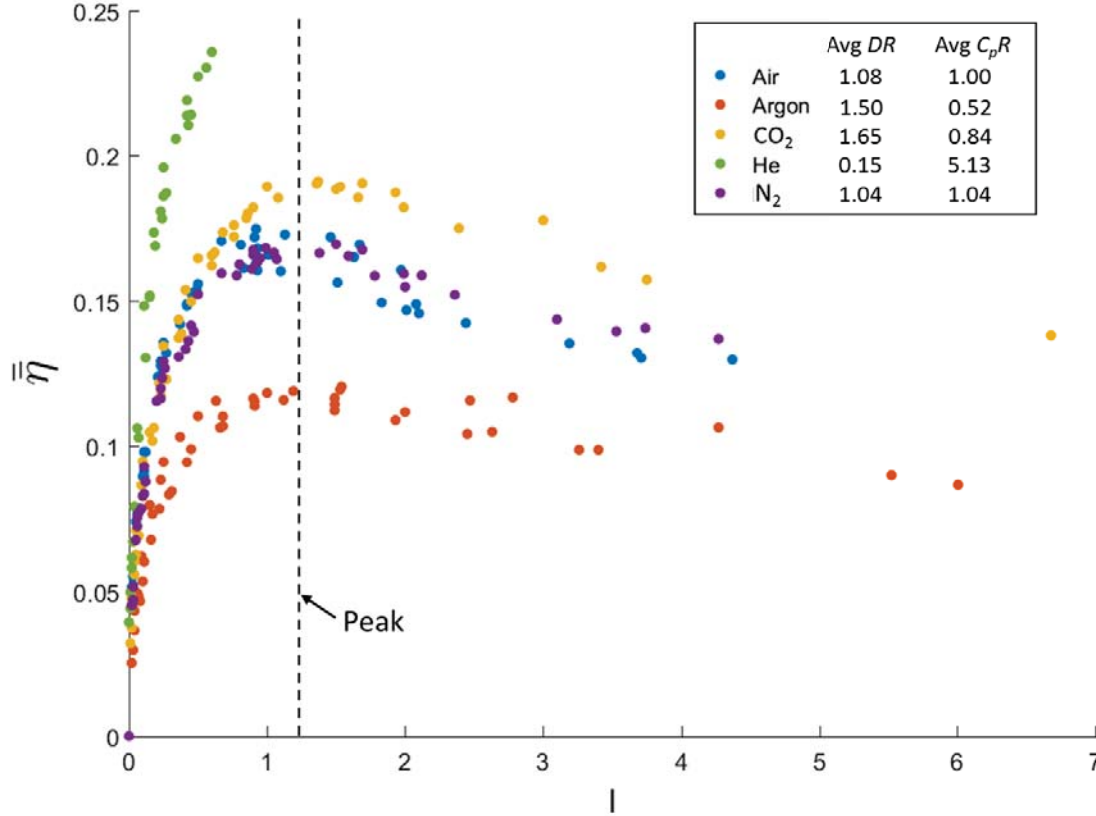


Figure 4.8. Area-averaged η values ($\bar{\eta}$) at all resulting I values. DR and C_pR values listed are averages among all test points.

4.2.3 Advective Capacity Ratio, ACR

As seen previously in the M and I results, the adiabatic effectiveness exhibits some dependence on C_pR . To determine the nature of this dependence, the matched ACR data was analyzed. First, the centerline η plots were generated and as seen in Figure 4.9, ACR was able to collapse the η profiles of all five gases when matching an ACR value of 0.25. The close grouping of profiles was seen to an even greater extent when the coolant flow rate was increased to $ACR = 0.5$ as seen in Figure 4.10. These

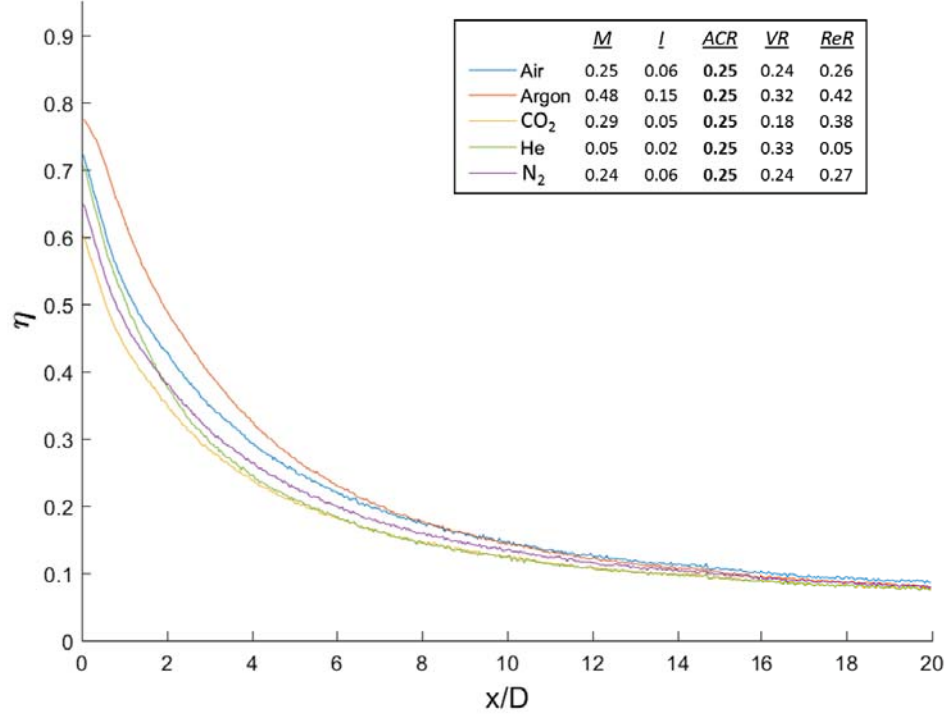


Figure 4.9. Distribution of η along $y/D=0$ at the matched $ACR = 0.25$ case.

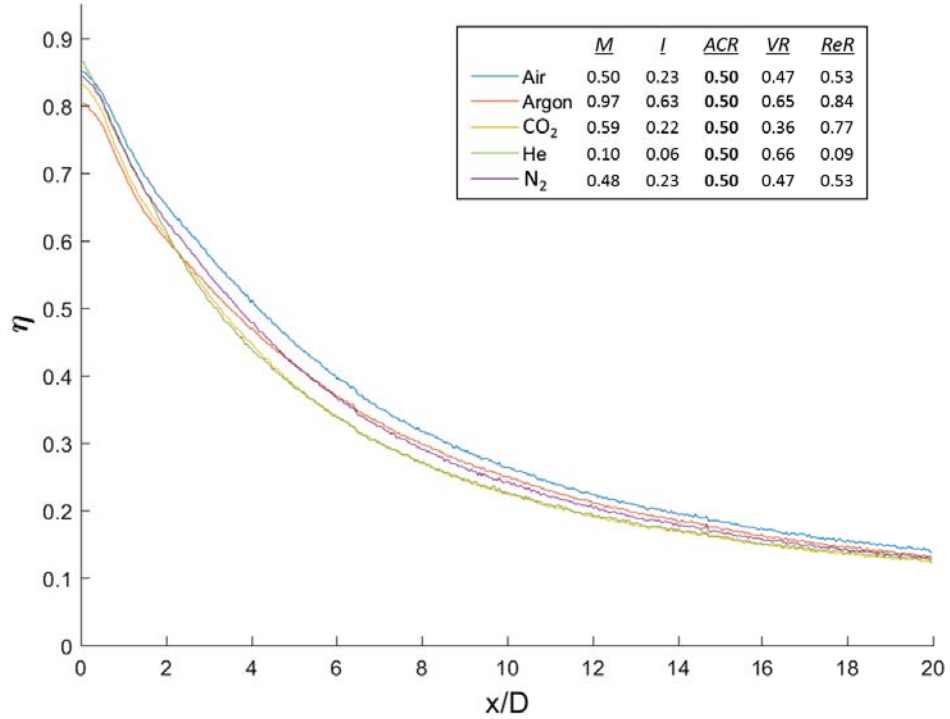


Figure 4.10. Distribution of η along $y/D=0$ at the matched $ACR = 0.50$ case.

results were something that has never been seen in any published literature. Despite having DR values ranging from approximately 0.15 to 1.65, the inclusion of the $C_p R$ allows ACR to account for each gas's ability to cool the surface, ultimately resulting in matched η profiles. Although the lines do not exactly lie on top of one another, which would indicate an exact scaling parameter, ACR still performed much better than previously seen from M or I experiments in Figures 4.3 and 4.6.

The grouping of η profiles spread as the flow rate was increased for higher ACR values. Figure 4.11 shows the $ACR = 1.5$ case where argon appears as the lowest η profile and helium the highest. Previously in the M and I results, if there was separation between the profiles, the lines on each plot would be somewhat ordered with increasing ACR . With the matched ACR lines, this order appeared to be in decreasing I values with helium having the lowest and argon the highest. All gases except for helium had an I value greater than the peak I of 1.2 from Figure

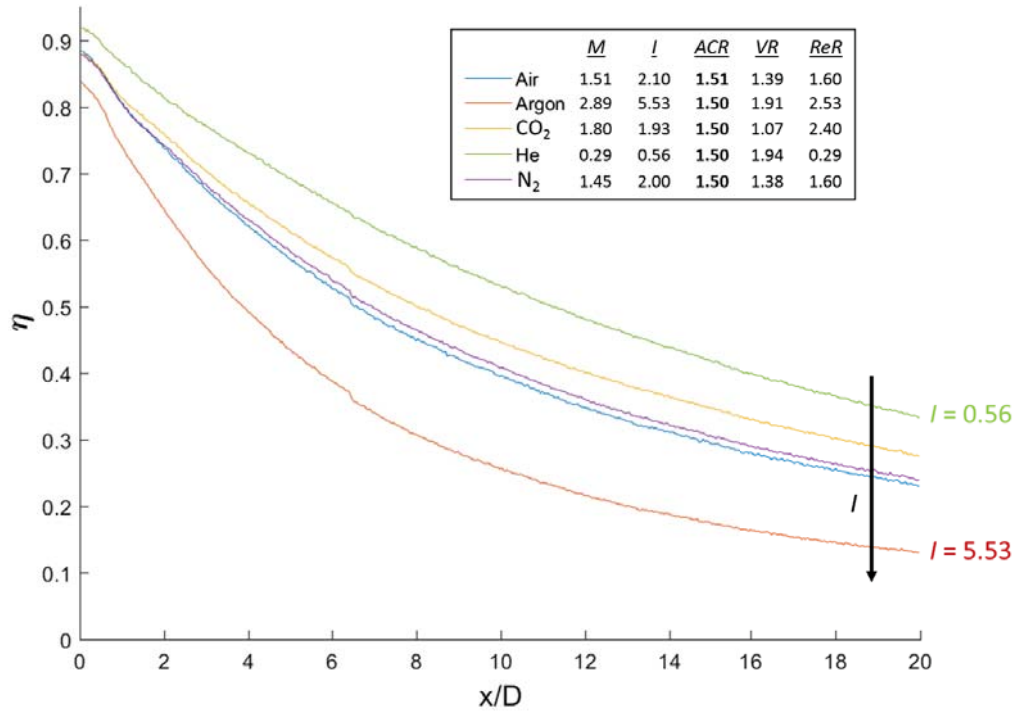


Figure 4.11. Distribution of η along $y/D=0$ at the matched $ACR = 1.50$ case.

4.8, leading to the conclusion that the separation of the *ACR* lines was due to jet separation effects. With its extremely high *I* value of 5.53, argon could be assumed to be the most separated which explained its relatively low η values. Helium's relatively high η profile is also explained by its low *I* value, indicating that the helium jet remained fully attached to the surface, and thus provided the best cooling.

The effects of jet separation were best visualized when the area-averaged η values were plotted against their respective *ACR* values, seen in Figure 4.12. This plot further illustrates *ACR*'s ability to scale η no matter the density ratio or specific heat ratio, as all gases initially follow the same path. Then, one by one, the distributions deviate from the general trend, and taper away similar to the *M* and *I* cases. The most interesting part about each deviation is that it occurred once the gas reached an *I* value of approximately 0.67. Each of the arrows seen in Figure 4.12 is placed at the *ACR* value where the *I* value for each gas reaches the value of 0.67. The limits of the current facility prevented helium testing of *ACR* values greater than 1.5, so the green arrow is the predicted deviation point at $I = 0.67$. Given this compelling evidence, it was concluded that *ACR* is the best scaling parameter for the adiabatic effectiveness for *I* values less than 0.67, where the coolant jet is assumed to be fully attached. The effects of jet separation reduce the accuracy of *ACR* as the complicated flow physics of a separated jet begin to dominate how the coolant is spread on the surface. It should be noted that this conclusion is limited to the tested configuration of a flat-plate geometry with an axial-oriented cooling hole. However, it is anticipated that similar results could be replicated with other shaped cooling hole geometries.

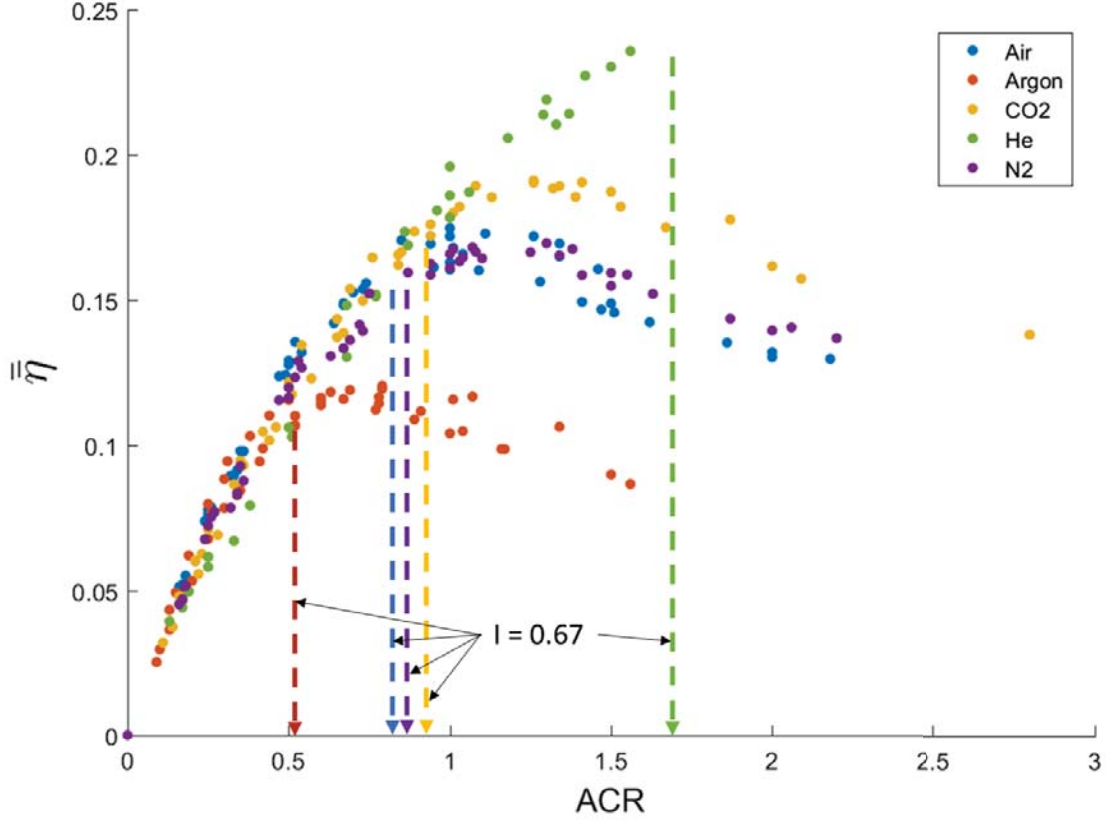


Figure 4.12. Area-averaged η values ($\bar{\eta}$) at all resulting ACR values. Deviation points at $I = 0.67$ labeled for each gas.

4.2.4 Velocity Ratio, VR

In an effort to fully examine all available coolant flow rate parameters, the data from the VR sweep was analyzed next. The ratio was not expected to accurately scale η because VR only accounts for the velocity of the coolant and freestream, allowing density and specific heat differences to influence the data. As seen in both Figure 4.13 and 4.14, matching VR yields similar results as matching M and does not scale between all five gases. Without considering the DR effects, VR also performs poorly with respect to the $\bar{\eta}$ distributions in Figure 4.15. The value of using multiple gases with varying properties was also apparent in the VR results, as one may conclude that VR performs well had the argon and helium data were removed.

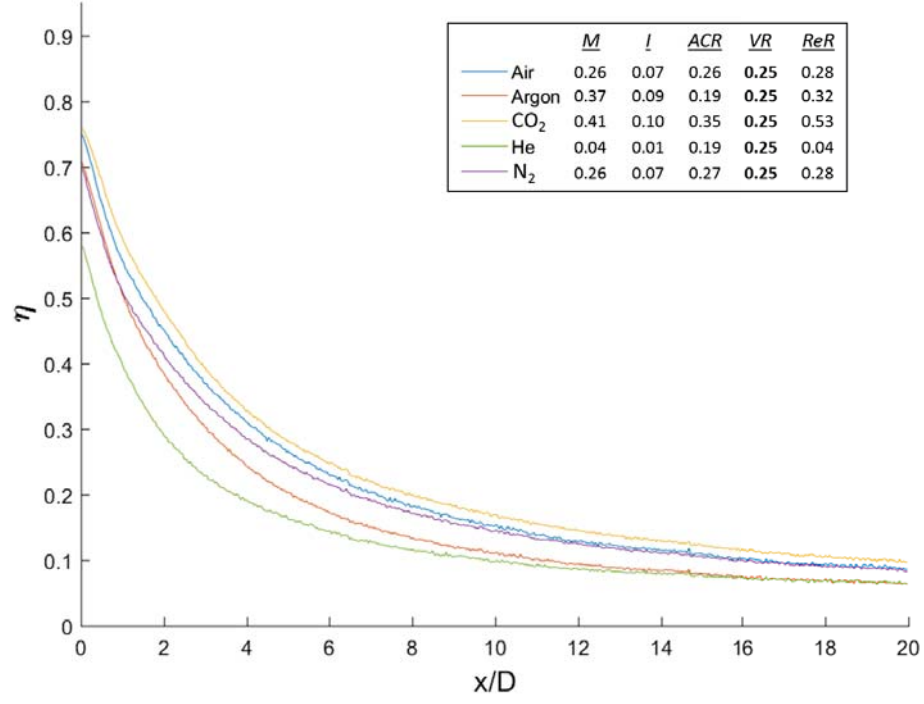


Figure 4.13. Distribution of η along $y/D=0$ at the matched $VR = 0.25$ case.

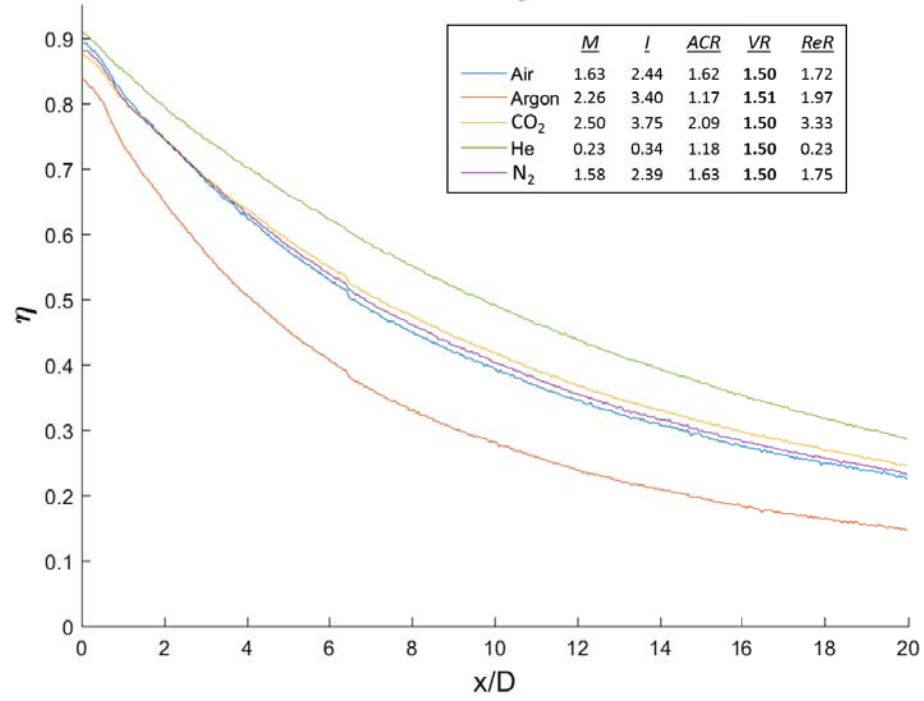


Figure 4.14. Distribution of η along $y/D=0$ at the matched $VR = 1.5$ case.

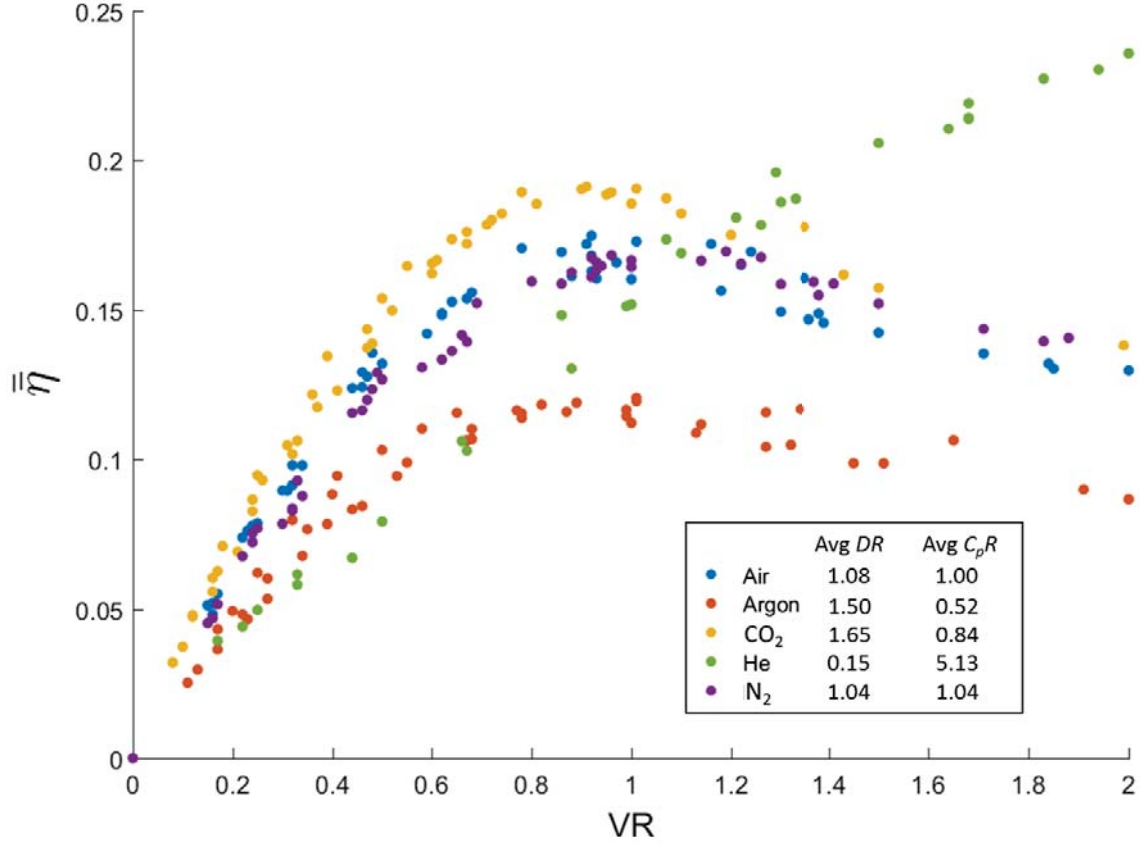


Figure 4.15. Area-averaged η values ($\bar{\eta}$) at all resulting VR values. DR and $C_p R$ values listed are averages among all test points.

4.2.5 Reynolds Number Ratio, ReR

The final coolant flow rate parameter to be analyzed was the Reynolds Number Ratio, which differs from the other parameters by including the dynamic viscosity ratio between the coolant and freestream gases. When analyzing the centerline η data in Figures 4.16 and 4.17 and the $\bar{\eta}$ distributions in Figure 4.18, the results showed a close resemblance to the M data at both flow rates. However, the data for the ReR cases showed an interesting trend when CO₂ was used as the coolant gas. Comparing the contour plots seen in Figures 4.19 and 4.20, it was noticed that the CO₂ coolant plume stayed the same width, while the coolant plumes of the three other gases narrowed. This was confirmed by plotting the lateral η distributions at

the $x/D=3$ location, seen in Figures 4.21 and 4.22. At the higher values of ReR , the coolant plume spread further laterally than the other three gases. The presence of this spreading phenomenon during CO_2 tests has been previously observed in the study done by Wiese et al. [17]. The authors were unable to attribute the spreading to a specific property or parameter, and the scope of the current study also prevented a solid conclusion from being made.

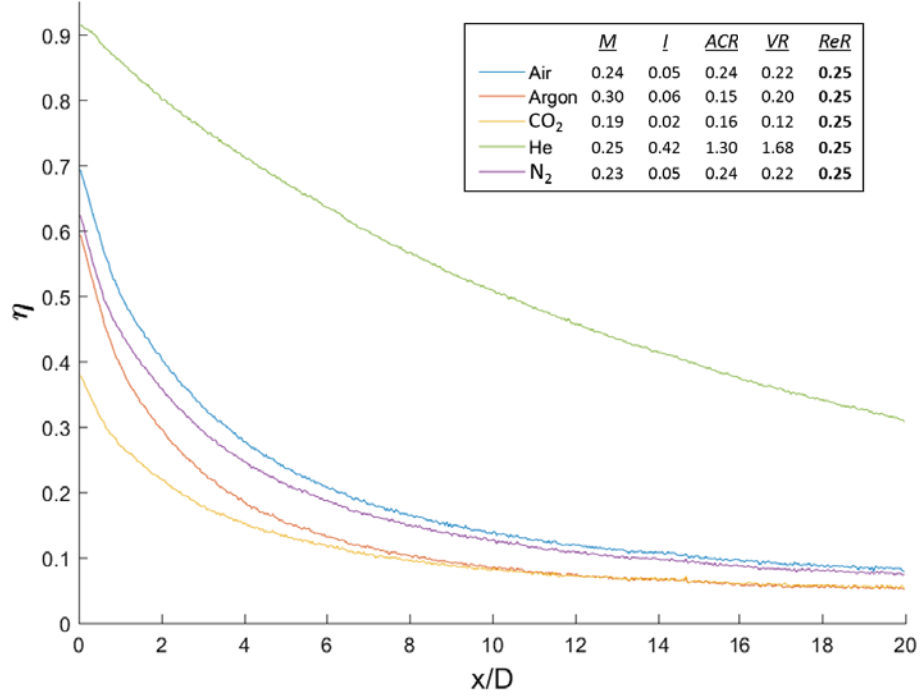


Figure 4.16. Distribution of η along $y/D=0$ at the matched $ReR = 0.25$ case.

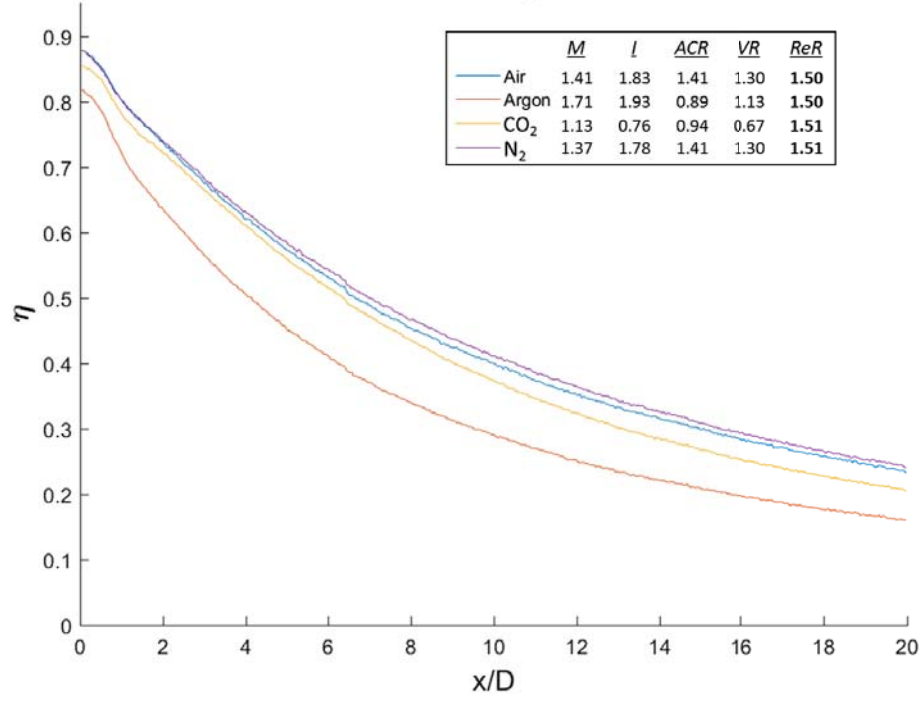


Figure 4.17. Distribution of η along $y/D = 0$ at the matched $ReR = 1.5$ case.

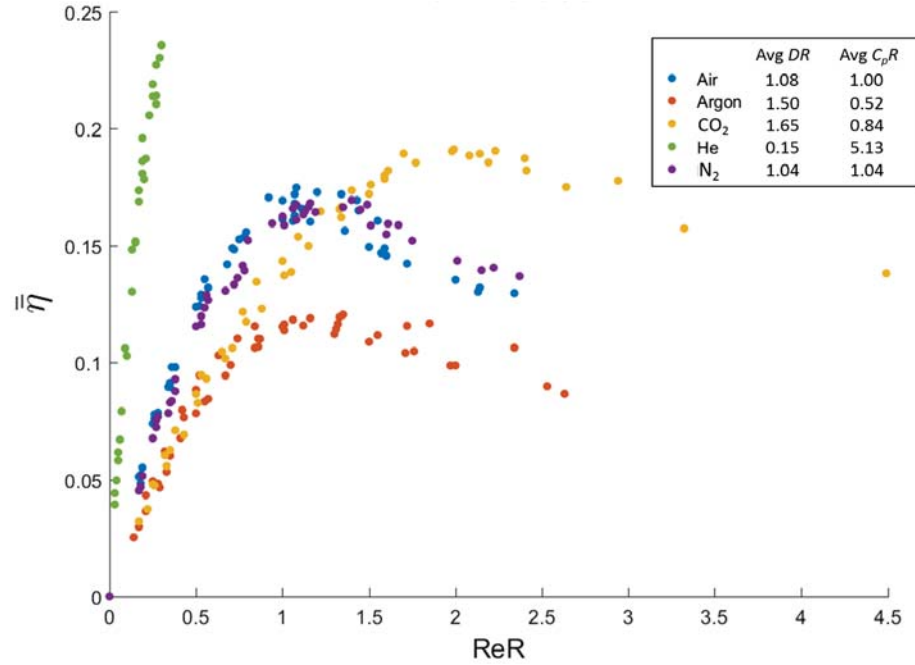


Figure 4.18. Area-averaged η values ($\bar{\eta}$) at all resulting ReR values. DR and $C_p R$ values listed are averages among all test points.

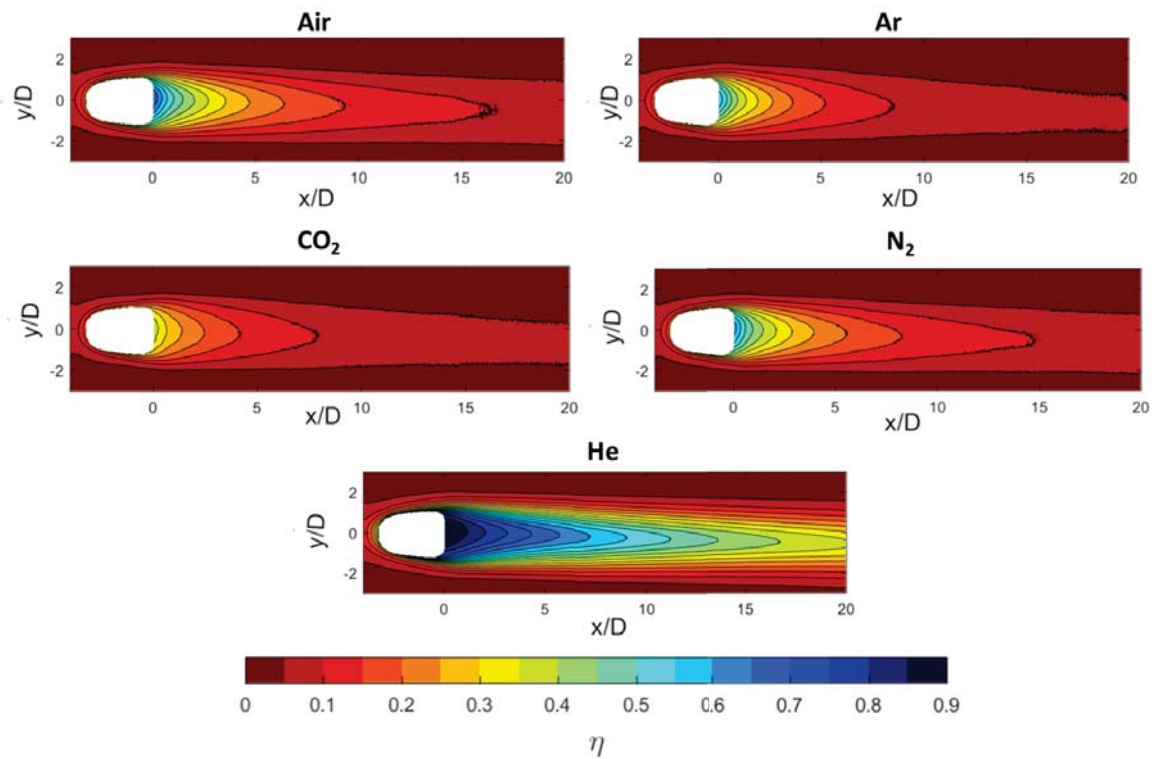


Figure 4.19. Contours of η at the matched $ReR = 0.25$ case.

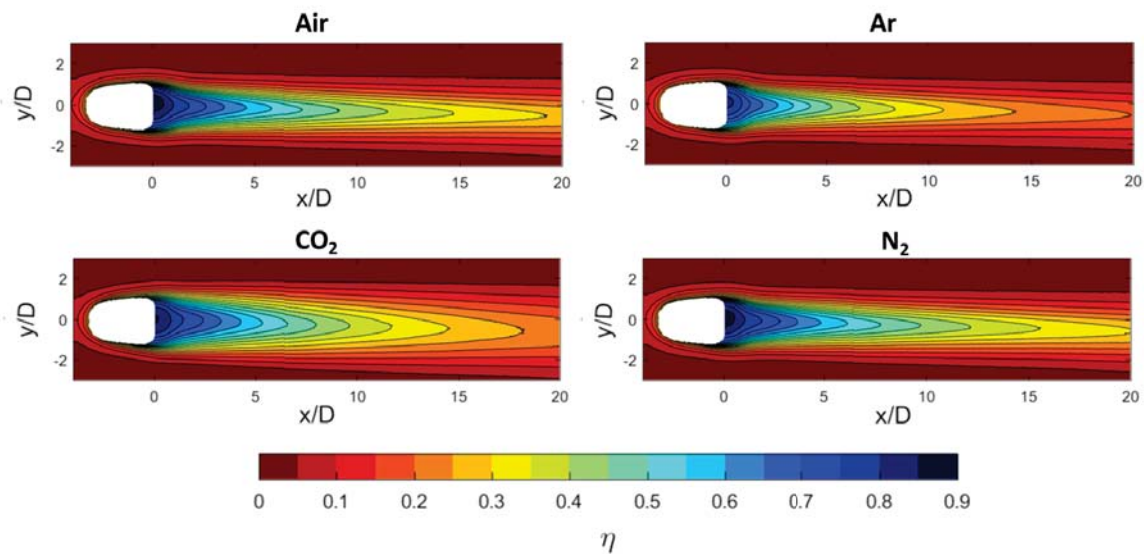


Figure 4.20. Contours of η at the matched $ReR = 1.5$ case.

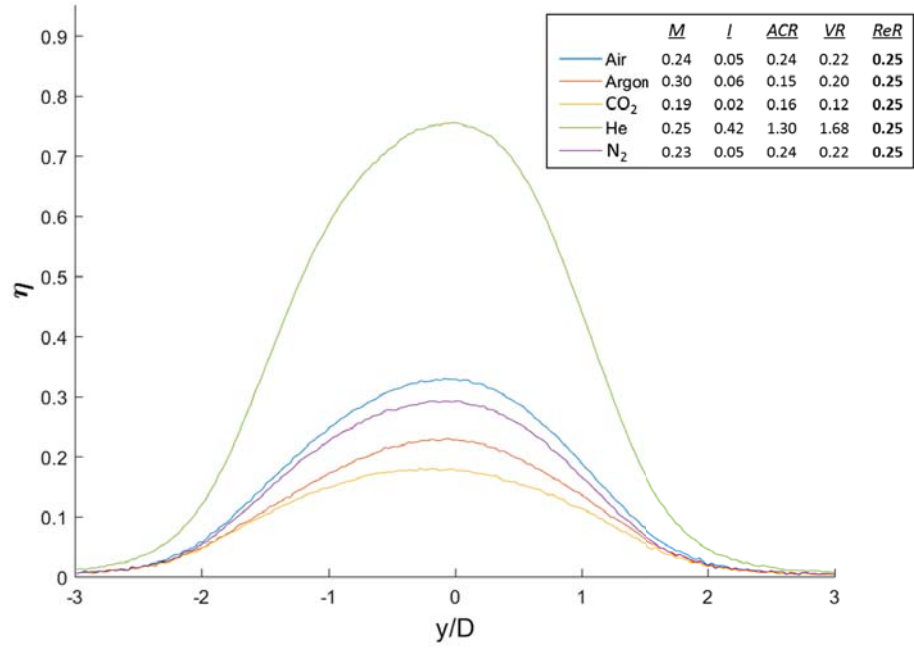


Figure 4.21. Distribution of η along $x/D=3$ at the matched $ReR = 0.25$ case.

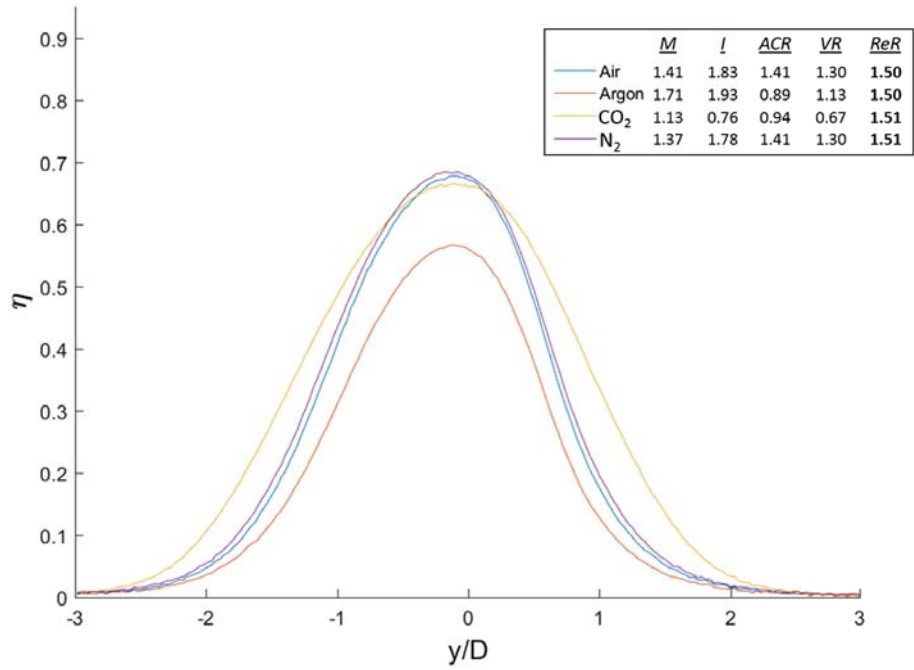


Figure 4.22. Distribution of η along $x/D=3$ at the matched $ReR = 1.5$ case.

4.3 Analysis of Previously Collected Data

The broad utility of using ACR in low-temperature scaling research is best seen by applying the ACR formula to previously conducted studies. For example, by analyzing the inconclusive results of the study conducted by Sinha et al. [14] using ACR , the variation between density ratios seen in Figure 4.23 is explained by varying ACR values. In the first $I = 0.2$ case, the two ACR values were calculated to be 0.49 for the $DR = 1.2$ case and 0.57 for the $DR = 1.6$ case. A higher ACR value explains the observed higher η values. In the second case, $I = 0.3$, where the two DR presented are 1.2 and 2.0, the difference between the presented η distributions is greater than in the $I = 0.2$ case, and this is mirrored by a larger difference between ACR values of 0.60 and 0.78 respectively. As seen in the current study, ACR cannot accurately predict η in regions where the jet has separated from the surface, as seen in the $I = 0.5$ case. The three ACR values for this case (in order of increasing DR) were calculated to be 0.77, 0.90, and 1.01. Starting at an X/D location of 1.5, the jet appears to reattach the the surface, and the η profiles become ordered again according to increasing ACR . If the authors had matched ACR , it is possible that the η profiles would have almost exactly matched in regions where the jet is attached to the surface.

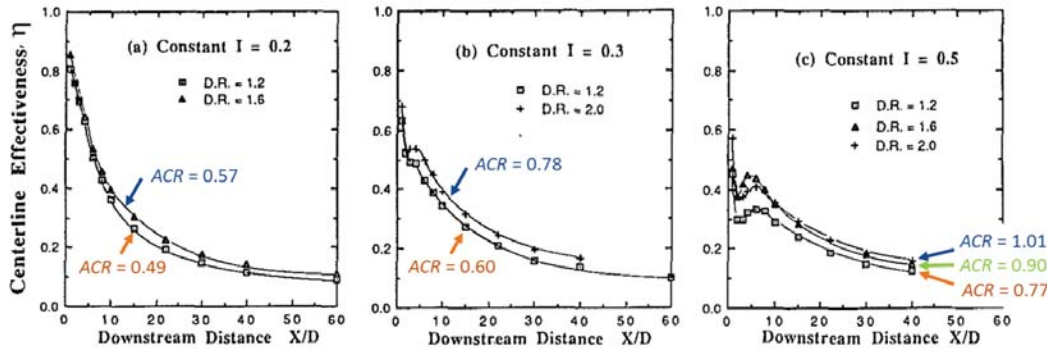


Figure 4.23. Matched I cases with varying DR with calculated ACR values. Adapted from [14].

When the authors conducted matched M cases at low flow rates ($M=0.5$), the η profiles matched well between the two presented DR values, seen in Figure 4.24. This result is also explained by ACR as the specific heat changes very little at low temperatures, as seen in Table 2.1. This results in calculated ACR values of 0.49 for $DR = 1.2$ and 0.51 for $DR = 2.0$, which explains the close matching of M with a fully attached coolant jet. Once the jet becomes fully separated, as in the $M = 1.0$ and $VR = 0.5$ cases, ACR does not explain the varying η values between density ratios, consistent with the findings in Section 4.2.3.

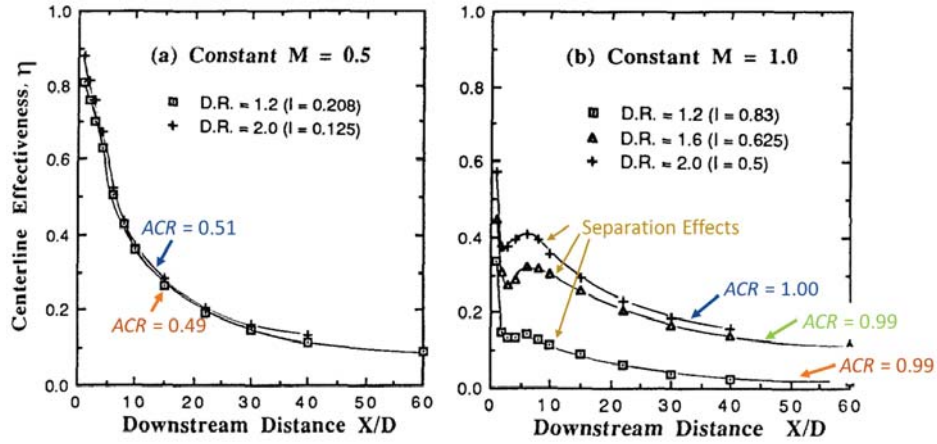


Figure 4.24. Matched M cases with varying DR with calculated ACR values. Adapted from [14].

As explained in Chapter 2, cylindrical holes are very prone to jet separation, which is why these effects were seen extensively in the Sinha et al. data. These effects are mitigated by use of shaped holes, which were used in the Schroeder and Thole study developing the 7-7-7 hole [13]. Since the current study utilized the 7-7-7 hole for testing, it was expected that differences in ACR before the separation limit of $I = 0.67$ should explain any variations seen in Figure 4.25. For the matched M cases, only the first two points had I values less than 0.67. The calculated ACR values for the $M = 0.5$ point were 0.50 for $DR = 1.2$ and 0.50 for $DR = 1.5$. The small difference between the measured values can be explained because the ACR

values are also matched at this point. This was also true at the $M = 1.0$ point where the calculated ACR values matched at 1.0. A similar trend was seen when calculating the ACR values for the I cases. The first two points on the matched I plot were approximately 0.1 apart, and the ACR values also slightly differed, resulting in $ACR = 0.49$ for $DR = 1.2$ and $ACR = 0.55$ for $DR = 1.5$. Even though the second point lies beyond $I = 0.67$, ACR still predicts the slight difference yielding values of 1.10 for $DR = 1.2$ and 1.23 for $DR = 1.5$. These results prove that ACR is a powerful tool for scaling η for flat plate geometries.

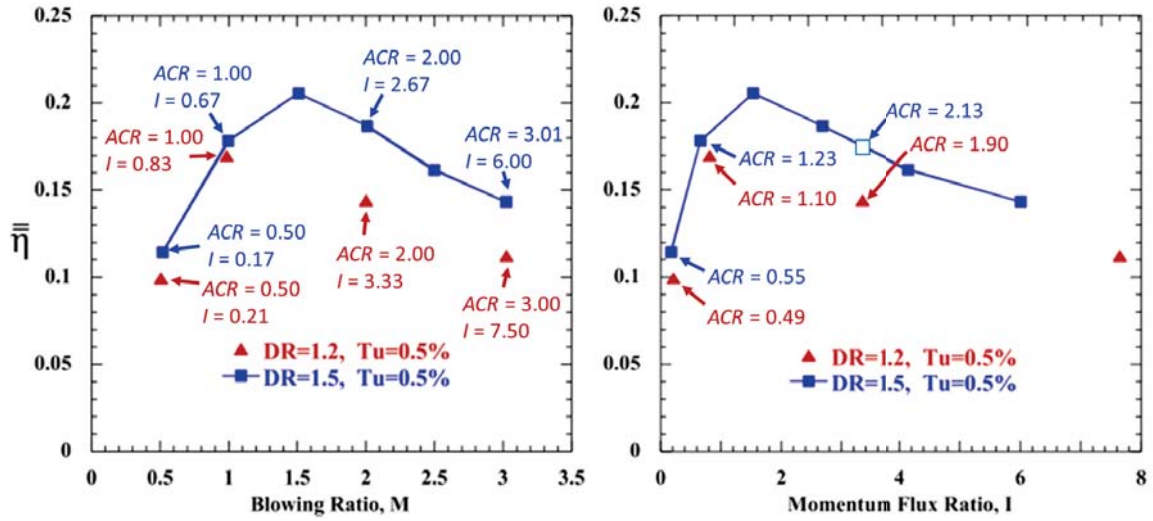


Figure 4.25. Area-averaged η at various M and I values. Averaged over $x/D = 3-35$. Adapted from [13].

V. Conclusion

5.1 Conclusions

The main objective of this research was to determine the best coolant flow rate parameter to use during low temperature testing to scale the adiabatic effectiveness. This objective was evaluated using five parameters and five gases with varying properties. One of the gases used was helium, which would provide compelling evidence for the use of one parameter over the others as it is drastically lighter than the freestream air. Prior testing in the current facility with a different geometry found the presence of temperature separation problems when using helium. This generated a secondary objective of determining the suitability of helium to be used on the new flat plate geometry.

5.1.1 Temperature Separation Results

Testing the secondary objective before the main objective ultimately proved that helium was indeed suitable for use on this geometry. Although temperature deviations were seen on the test surface when the helium coolant was run at approximately the same temperature as the freestream flow, the magnitude of these deviations was $\pm 3K$. Additionally, these maximum deviations affected very small areas in regions downstream of the cooling hole's upper and lower edges. The maximum deviation affecting a larger area along the centerline was actually on the order of $\pm 1.5K$. During normal testing, a temperature difference between the freestream and coolant was approximately 30K, so a 1.5K temperature deviation would result in approximately a 0.05 change in the resulting η distributions. This effect on η was deemed small enough to proceed with helium testing. During data analysis, efforts were made to either avoid the areas prone to temperature separation, or minimize their effects by

averaging over a large area.

5.1.2 Scaling Results

Through testing all five available coolant flow rate parameters, M , I , ACR , VR , and ReR , this study provided comprehensive results for scaling with an axially-oriented, shaped cooling hole on a flat plate. First, it was confirmed that I best predicts jet separation, even with the shaped cooling hole geometry. For this particular geometry, peak area-averaged η values occur at a value of $I \approx 1.2$. This result was consistent with many prior film cooling studies. The second result from this research is that ACR will almost exactly scale adiabatic effectiveness results, as long as the cooling jet remains fully attached with I values less than 0.67. Once the jet detaches, the complicated flow physics of separated cooling jets begin to dominate how the coolant is spread onto the test surface. A result this definitive has never been seen in any prior scaling study with most having to define regions where M is preferable versus where I is preferable. By including the specific heat of both the coolant and freestream, ACR is the only quantity needed to accurately quantify the cooling capacity of any given gas, even those with drastically different properties such as with helium. This result makes sense as the perfect scaling parameter would quantify how much cooling the coolant flow is capable of. The cooling process is governed by thermal diffusion, and one of the main driving factors in thermal diffusion is the specific heat. Matching the specific heat ratio results in good prediction of thermal diffusion on the flat plate surface, and thus a good prediction of η values. Even once the jet separates from the surface, ACR can still be used to predict which coolant flows will perform better than others.

5.2 Significance of Research

The ability of a single coolant flow rate parameter to almost exactly scale η in any region has far reaching benefits for many different communities. For the academic community, these results prove the importance of the specific heat and the specific heat ratio on scaling η . Decades of research have been conducted searching for a way to make either M , I , or VR scale η across multiple density ratios, and this research proves that if c_p is accounted for, there is no need to match the density ratio. This is not to say that the density ratio is not important in film cooling, as many studies have proven density ratio effects, but when used in conjunction with the specific heat ratio, accurate η predictions can be made without exactly matching the density ratio of an actual engine.

The ability to accurately scale low temperature results also allows for cheap and simple testing of new geometries and configurations. By matching ACR during testing, η results can be scaled up to engine operating conditions and confidently predict what the adiabatic wall temperature of each turbine component will be. With increased confidence, engineers would be able to lower safety factors built into each turbine blade, resulting in potential weight and efficiency savings. Any reduction of weight or increase in efficiency would reduce the operating cost of the gas turbine engine, allowing the Department of Defense to spend less on fuel and more on developing new technologies.

5.3 Recommendations for Future Work

There are multiple directions for this research to continue. One of these is to conduct similar pressure sensitive paint studies on the same geometry. The surface of the foam insert used in this study was coated with epoxy prior to testing in order to provide for an easy transition to these experiments. This would provide further

insight into the differences between the methods, and also test *ACR*'s scaling ability between multiple different measurement techniques. As seen earlier, PSP tends to over-predict η values as it does not account for thermal diffusion between the coolant gas and the freestream flow. The use of *ACR* is not expected to improve the PSP results, as *ACR* relies on thermal diffusion to predict η values. Another direction is testing overall effectiveness (ϕ) on flat plate geometries using matched Biot number models. The models could come in many forms with multiple hole layouts, different materials, and multiple manufacturing techniques.

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1. REPORT DATE (DD-MM-YYYY) 22-03-2018		2. REPORT TYPE Master's Thesis			3. DATES COVERED (From — To) January 2017 — March 2018	
4. TITLE AND SUBTITLE Influence of Coolant Flow Rate Parameters in Scaling Gas Turbine Cooling Effectiveness on a Flat Plate				5a. CONTRACT NUMBER		
				5b. GRANT NUMBER		
				5c. PROGRAM ELEMENT NUMBER		
				5d. PROJECT NUMBER		
6. AUTHOR(S) Fischer, Jacob P., Second Lieutenant, USAF				5e. TASK NUMBER		
				5f. WORK UNIT NUMBER		
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) Air Force Institute of Technology Graduate School of Engineering and Management (AFIT/ENY) 2950 Hobson Way, Building 640 WPAFB OH 45433-7765					8. PERFORMING ORGANIZATION REPORT NUMBER AFIT-ENY-MS-18-M-256	
9. SPONSORING / MONITORING AGENCY NAME(S) AND ADDRESS(ES) Air Force Research Lab, Turbine Engine Division, Turbomachinery Branch Dr. Andrew T. Lethander 1950 5th St WPAFB OH 45433 (937) 255-6779 andrew.lethander@us.af.mil					10. SPONSOR/MONITOR'S ACRONYM(S) AFRL/RQTT	
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14. ABSTRACT Effective film cooling relies on the performance of the selected cooling hole shape and placement of the holes on the surface of the blade. To test each holes effectiveness, large scale, low-temperature testing has been used to reduce cost and complexity. The resulting quantity from this testing is called the adiabatic effectiveness (η). The adiabatic effectiveness can then scaled up to engine operating conditions to predict how the selected film cooling hole will perform. This process is complicated by the nonlinear variation of gas properties between coolant and freestream conditions as well as between the experimental and engine conditions. In this study, various gases were used as coolant to simulate changes in gas properties as one might have between experimental and engine conditions. It was found that when specific heat is included in the form of a new parameter known as the Advective Capacity Ratio (ACR), adiabatic effectiveness results scale well between different cooling gases. Specifically at low momentum flux ratios, use of ACR produced very similar adiabatic effectiveness profiles between all gases.						
15. SUBJECT TERMS Film Cooling, Scaling						
16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON	
a. REPORT	b. ABSTRACT	c. THIS PAGE			James L. Rutledge, AFIT/ENY	
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